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# Direct-Contact Condensation of Steam on Cold Water in Stratified Countercurrent Flow

ORIGINAL

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Prepared by S. G. Bankoff, H. J. Kim

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Prepared for  
U.S. Nuclear Regulatory  
Commission

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## ABSTRACT

Local condensation heat transfer coefficients have been measured for countercurrent stratified flow of steam and cold water at atmospheric pressure in a flat plate geometry at an inclination of  $4^\circ$ ,  $30^\circ$ , and  $33^\circ$  from the horizontal. Pressure gradients are also measured which allow the calculation of local interfacial shear stresses. In addition, interfacial wave amplitudes are measured locally by means of contact probes, from which wave frequencies and spectral analyses can be deduced. The data are correlated in terms of the gas and liquid Reynolds numbers, both for the Nusselt number and dimensionless wave amplitudes. An improved correlation is obtained by employing the wave amplitudes directly in the heat transfer correlation. Shear stresses are calculated, which allow a new correlation for the interfacial friction factor. A turbulence-centered model is also modified, giving good agreement with the data.

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# NOMENCLATURE

|                                                                      |                                                           |
|----------------------------------------------------------------------|-----------------------------------------------------------|
| B                                                                    | Channel width                                             |
| b                                                                    | Magnitude of $y^+$ at interface, $u_* \cdot \delta / \nu$ |
| Cp                                                                   | Specific heat                                             |
| C, C <sub>1</sub> , C <sub>2</sub>                                   | Constants                                                 |
| D                                                                    | Tube diameter                                             |
| D <sub>L</sub>                                                       | Molecular diffusivity of liquid                           |
| f                                                                    | Friction coefficient                                      |
| F( $\alpha_e$ )                                                      | Weakly varying function given by Ref. (21)                |
| g                                                                    | Acceleration of gravity                                   |
| H                                                                    | Channel height                                            |
| h                                                                    | Local heat transfer coefficient                           |
| ifg                                                                  | Latent heat of evaporation                                |
| k                                                                    | Thermal conductivity                                      |
| k <sub>L</sub>                                                       | Mass transfer coefficient                                 |
| L                                                                    | Length of test section                                    |
| m                                                                    | Mass flow rate                                            |
| N <sub>f</sub>                                                       | Frequency of crossings                                    |
| Nu                                                                   | Nusselt number, $h \cdot \delta / k$                      |
| Nut                                                                  | Turbulent Nusselt number, $h \cdot \lambda_t / k$         |
| n <sub>1</sub> , n <sub>2</sub> ,<br>n <sub>3</sub> , n <sub>4</sub> | Constants                                                 |
| P                                                                    | Static pressure, Probability                              |
| Pr                                                                   | Prandtl number                                            |
| Q                                                                    | Volumetric flow rate per unit width                       |
| q                                                                    | Heat flux                                                 |
| Re                                                                   | Reynolds number, $W / \mu$                                |
| Ret                                                                  | Turbulent Reynolds number, $u_t \cdot \lambda_t / \nu$    |
| S                                                                    | Height of water layer                                     |
| So                                                                   | Particular position of water layer                        |
| S <sub>1</sub>                                                       | Mean wave amplitude                                       |
| S <sub>2</sub>                                                       | R.M.S. wave amplitude                                     |
| St <sub>t</sub>                                                      | Turbulent Stanton number                                  |
| T                                                                    | Temperature                                               |
| t                                                                    | Time                                                      |
| t <sub>0</sub>                                                       | Time interval                                             |
| t <sub>exp</sub>                                                     | Exposure time                                             |
| u                                                                    | Velocity                                                  |
| u <sub>1</sub>                                                       | Velocity at $y_1$                                         |
| u <sub>2</sub>                                                       | Velocity at $y_2$                                         |
| u <sup>+</sup>                                                       | Dimensionless velocity scale, $u / u_*$                   |
| u <sub>t</sub>                                                       | Turbulent velocity scale                                  |
| u <sub>*</sub>                                                       | Shear velocity                                            |
| W                                                                    | Mass flow rate per unit width                             |
| X                                                                    | Parameter                                                 |
| x                                                                    | Coordinate of steam flow direction                        |
| y                                                                    | Coordinate normal to flow                                 |
| y <sub>1</sub>                                                       | Thickness of viscous sublayer                             |
| y <sub>2</sub>                                                       | The point of zero shear stress                            |
| y <sup>+</sup>                                                       | Dimensionless length scale, $u_* y / \nu$                 |
| z                                                                    | Distance from water inlet                                 |

# GREEK SYMBOLS

|             |                                                                                  |
|-------------|----------------------------------------------------------------------------------|
| $\alpha$    | Void fraction                                                                    |
| $\alpha_e$  | Dimensionless exposure time for a single eddy, $t_{exp} \cdot u_t / \lambda_t$   |
| $\beta$     | Ratio of gravity force to wall shear stress                                      |
| $\gamma$    | Ratio of gravity force to interfacial shear stress                               |
| $\delta$    | Mean water layer thickness                                                       |
| $\delta'$   | Dimensionless water layer thickness, $\delta \cdot (v^2 / g \sin \theta)^{-1/3}$ |
| $\epsilon$  | Dissipation rate per unit mass                                                   |
| $\theta$    | Inclination angle in degree                                                      |
| $\kappa$    | Von Karman constant                                                              |
| $\lambda_t$ | Turbulent length scale                                                           |
| $\mu$       | Viscosity                                                                        |
| $\nu$       | Kinematic viscosity                                                              |
| $\rho$      | Density                                                                          |
| $\sigma$    | Surface tension                                                                  |
| $\tau$      | Shear stress                                                                     |
| $\phi$      | Reynolds momentum flux                                                           |

## SUBSCRIPTS

|     |                      |
|-----|----------------------|
| a   | Adiabatic            |
| b   | Upper wall surface   |
| c   | Condensation         |
| D   | Hydraulic diameter   |
| E   | Eddy                 |
| g   | Vapor                |
| i   | Interface            |
| in  | Inlet                |
| l   | Liquid               |
| max | Maximum              |
| o   | Liquid wall surface  |
| r   | Reference            |
| sat | Saturation           |
| t   | Turbulent            |
| x   | Steam flow direction |
| z   | Water flow direction |

## SUPERSCRIPTS

|   |                           |
|---|---------------------------|
| - | Average                   |
| * | Presence of mass transfer |
| ' | Dimensionless             |

## 1. INTRODUCTION

In a small-break loss-of-coolant accident (LOCA) in a PWR, with the main pumps inoperative, the primary side of the steam generators can go into the reflux condenser mode, even before the primary pressure falls substantially below the secondary system pressure. Condensate then drains back through the hot legs towards the reactor. At about the same time the emergency core coolant (ECC) water is injected into the hot legs. Hence a countercurrent stratified flow of steam and subcooled water may be set up within the hot leg(s). If the differential pressure between the reactor upper plenum and the cold legs can be kept sufficiently low, the core will not uncover. This is a consequence most devoutly to be desired, as the Three Mile Island experience showed. The upper plenum pressure is determined by the balance between the rate of production of steam in the overheated core, and the rate of condensation in the steam generators, hot legs and downcomer.

Other locations where steam and cold water come into contact in the large-break accident are the downcomer, mixing tee and upper plenum. Available data for direct-contact local condensation in countercurrent flow involving fairly thick layers of cold water are quite limited. The objective of the present work is to study the dependence of condensation rates, interfacial wave structure and interfacial shear stress on the water flow rate, water temperature, aspect ratio of the test section and inclination of the test section.

## 2. REVIEW OF PREVIOUS WORK

Film condensation has been studied by many investigators, beginning with Nusselt<sup>(1-10)</sup>. However, condensation on thick layers of cold water, where heat transport to the solid boundary is either absent or negligible, has been much less well studied. Linehan<sup>(11)</sup> made measurements of local water temperatures at the wall, and mean film thicknesses, for a cocurrent stratified flow of saturated steam over a subcooled water film in a rectangular channel. Based on his data, he proposed that the interfacial shear stress in the presence of condensation is simply augmented by an amount equal to the condensation rate times the average vapor velocity. Thus the Stanton number is constant and had the value 0.0073 for his data. Somewhat more elaborate results had been previously obtained by Silver and Wallis<sup>(12)</sup> and by Mickley, et al<sup>(13)</sup>.

In a rather different geometry consisting of a high speed jet in contact with a co-current coaxial flow of subcooled water, Young, et al.<sup>(14)</sup> obtained the correlation

$$St = 6.5 Re_D^{-0.40} \quad (2-1)$$

where the characteristic velocity was the interfacial velocity calculated from a Reynolds flux model. Theofanous<sup>(15)</sup> examined condensation data from a variety of sources for turbulent water jets and found that the assumption of a constant Stanton number correlated the data quite well. In the closest previous work to the present study, Segev, et al.<sup>(16)</sup> examined a stratified counter-current flow of steam and cold water in an inclined rectangular test section. When no bypass occurs (which is the flow regime of the present study), their data in the region of pebbly surface and roll waves were correlated by

$$Nu = 1.16 \times 10^{-3} Re_g^{0.28} Re_l^{0.87} Pr^{0.05} \quad (2-2)$$

for a fixed inlet water temperature of 23°C. However, upon varying the inlet water subcooling, a somewhat different correlation was obtained:

$$Nu = 8.5 \times 10^{-4} Re_g^{0.25} Re_l^{0.85} Pr^{0.5} \quad (2-3)$$

Lim, et al<sup>(17)</sup> measured condensation rates for cocurrent stratified flow of steam and cold water in a horizontal channel with the results:

$$St = 0.0344 Re_{g,x}^{0.58} Re_{l,x}^{-0.58} Pr^{-0.7} \quad (2-4)$$

for rough interface regime

$$St = 0.63 Re_{g,x}^{0.58} Re_{l,x}^{-0.91} Pr^{-0.7} \quad (2-5)$$

for smooth interface regime

which contrasts with the constant Stanton number result of Linehan.

Since the principal resistance to heat transfer is in this case in the liquid boundary layer under the interfacial waves, it is also useful to examine semi-theoretical correlations based upon liquid turbulent quantities. A beginning has been made in application of turbulent transport theory to the prediction of condensation heat transfer coefficients. In particular, the heat transfer analogy of the Brumfield, et al.<sup>(18)</sup> mass transport equation, based on free surface gas absorption data<sup>(21)</sup>, was reasonably successful in predicting average condensation coefficients for several sets of horizontal open channel flow<sup>(19,20)</sup>.

A few studies have also been made on the interfacial wave structure and its effect on the transport mechanism<sup>(22-25)</sup>. Telles, et al.<sup>(26)</sup> and Chu, et al.<sup>(27,28)</sup> obtained statistical data on waves in both vertical downward and horizontal flow. There was a random interfacial structure with a bimodal distribution, consisting of capillary waves on the base film flow, and solitary roll waves. Suzuki, et al.<sup>(29)</sup> measured the variation of the maximum height of the wavy liquid films as a function of the gas flow rate. They showed that the wave frequency was approximately proportional to the flow rate of liquid phase per unit width, irrespective of liquid viscosity or surface tension. Akai, et al.<sup>(30,31)</sup> correlated the interfacial wave structure with the flow properties in horizontal cocurrent stratified flow of air and water and also air and mercury. They employed a two-equation turbulence model which, when solved numerically, gave good agreement with their data and with their measurements of the liquid turbulence. This latter work seems promising for further development.

### 3. ANALYTICAL APPROACH

#### 3-1. Heat Transfer Coefficient

Consider a stratified steam-water flow in a plane geometry inclined at an angle  $\theta$  to the horizontal, with the following assumptions: i) steam is everywhere saturated, ii) the pressure of the steam is everywhere nearly constant, iii) both lower and upper walls are adiabatic, iv) two dimensional effects (in the cross-stream direction) are absent. The mass balance for the control volume 1) and 2) gives, as shown in Fig. 1:

$$dW_l = -dW_g \quad (3-1-1)$$

Upon considering energy balances for the control volume 1) and 2);

$$W_l C_p dT + dW_l C_p (T_l - T_r) + \{i_{fg} + C_p (T_g - T_r)\} dW_g = 0 \quad (3-1-2)$$

where mechanical energy terms have been neglected and  $T_r$  is an arbitrary reference temperature. Upon employing the mass balance, equation (3-1-1), and integrating from  $z=0$  (water inlet) to  $z$ , one obtains;

$$W_l = W_{l,in} \frac{i_{fg} + C_p (T_g - T_{l,in})}{i_{fg} + C_p (T_g - T_l)} \quad (3-1-3)$$

or

$$T_l = T_{l,in} + \frac{(W_l - W_{l,in}) \{i_{fg} + C_p (T_g - T_{l,in})\}}{C_p W_l} \quad (3-1-4)$$

Assuming the interface to be always at  $T_{sat}$ , the heat transferred to the liquid is related to the temperature gradient at the interface times an effective turbulent thermal conductivity:

$$q_c = -k_E \left| \delta \frac{\partial T_l}{\partial y} \right|_{\delta} = i_{fg} \frac{dW_l}{dz} \quad (3-1-5)$$

The condensation heat transfer coefficient is defined as:

$$q_c = h (T_g - T_l) = i_{fg} \frac{dW_l}{dz} \quad (3-1-6)$$

whence

$$h = \frac{i_{fg}}{(T_g - T_l)} \frac{dW_l}{dz} \quad (3-1-7)$$

The axial-average heat transfer coefficient is obtained by integrating from  $z=0$ :

$$\bar{h}_z = \frac{1}{z} \int_0^z h dz = \frac{i_{fg}}{z} \int_{W_{l,in}}^{W_l} \frac{1}{T_g - T_l} dW \quad (3-1-8)$$

Upon substituting equation (3-1-4) into equation (3-1-8) and performing the integration, one obtains:

$$\bar{h}_z = \frac{C_p}{z} \left[ (W_{l,in} - W_l) + W_{l,in} \left\{ 1 + \frac{C_p(T_g - T_{l,in})}{i_{fg}} \right\} \right. \\ \left. \times \ln \left\{ \frac{W_{l,in} \frac{C_p(T_g - T_{l,in})}{i_{fg}}}{W_{l,in} \left\{ 1 + \frac{C_p(T_g - T_{l,in})}{i_{fg}} \right\} - W_l} \right\} \right] \quad (3-1-9)$$

This equation relates the average heat transfer coefficient to the local liquid flow rate, and hence to the total condensation rate from the inlet to the point in question.

### 3-2. Interfacial Shear Stress

To evaluate the interfacial shear stress, it is assumed that i) the flow is steady, incompressible and non-entraining, ii) the actual time-varying water layer thickness is replaced with a temporal mean thickness, iii) the temperature of vapor is saturated and remains constant and iv) the pressure variation across the flow direction is negligible.

A momentum balance for the control volume on the steam side, shown in Fig. 2, shows that:

$$\tau_i = - \left\{ \frac{d}{dx} \int_0^H \rho_g u_g^2 dy - u_i \frac{dW_g}{dx} \right\} - (H-\delta) \frac{dP_g}{dx} - \rho_g \cdot g(H-\delta) \sin \theta - \tau_b \quad (3-2-1)$$

where the interfacial shear stress,  $\tau_i$ , has been expressed in terms of the condensation momentum flux, pressure gradient, hydrostatic and upper wall shear stress terms, respectively. Hanratty, et al.<sup>(32)</sup> found that, for the fully developed flow of a horizontal, stratified, turbulent air-water flow, the velocity profile in the gas phase near the wall was unaffected by the presence of the moving liquid phase. It is not clear that the same simplification can be applied to a condensing flow. Nevertheless, we shall make the same assumption here. The wall shear stress may then be calculated from the Blasius equation for the friction factor in turbulent flow, which is defined as:

$$f_b = 0.079 \text{ Re}_{g,D}^{0.25} \quad (3-2-2)$$

$$\tau_b = f_b \rho_g \cdot \bar{u}_g^2 / 2 \quad (3-2-3)$$

For cocurrent air-water flow in a horizontal channel, Cohen<sup>(33)</sup> found that the interfacial friction factor could be correlated with the liquid Reynolds number only:

$$f_i = 1.875 \times 10^{-5} \text{ Re}_l + 0.0068 \quad \text{for } \text{Re}_l < 340 \quad (3-2-4)$$

$$f_i = 0.23 \times 10^{-5} \text{ Re}_l + 0.0131 \quad \text{for } \text{Re}_l > 340 \quad (3-2-5)$$

$$\text{where } f_i = \frac{\tau_i}{\rho_g \cdot \bar{u}_g^2 / 2} \quad (3-2-6)$$

On the other hand, Bharathan, et al.<sup>(34)</sup> correlated their interfacial friction factor data for air-water countercurrent flow in a vertical tube by:

$$f_i = 0.005 + C_1 (D'/2)^{C_2} (1 - \sqrt{\alpha})^{C_2} \quad (3-2-7)$$

$$\text{where } \log_{10} C_1 = -0.56 + 9.07/D' \quad (3-2-8)$$

$$C_2 = 1.63 + 4.74/D' \quad (3-2-9)$$

$$\text{and } D' = D / \left\{ \frac{\sigma}{g \cdot (\rho_l - \rho_g)} \right\}^{\frac{1}{2}} \quad (3-2-10)$$

is the square root of a Bond number.

The effect of interfacial mass transfer on the frictional characteristics has been studied by several investigators. Silver and Wallis<sup>(12)</sup> use the Reynolds flux concept, arriving at:

$$f_i^*/f_i = \exp(\phi/2f_i) - \phi/f_i \quad (3-2-11)$$

$$\text{where } \phi/f_i = 2/(\rho_g \bar{u}_g f_i) dW_g/dx \quad (3-2-12)$$

can be looked upon as the ratio of the Reynolds momentum flux to the interfacial shear stress without condensation. An alternative expression, due to Mickley, et al.<sup>(13)</sup>, who employed stagnant-film theory, is:

$$f_i^*/f_i = (\phi/f_i) / \{\exp(\phi/f_i) - 1\} \quad (3-2-13)$$

However, after examining these relations, Linehan<sup>(11)</sup> proposed the simple relationship:

$$f_i^*/f_i = 1 - (\phi/f_i) \quad (3-2-14)$$

which is equivalent to

$$\tau_i = f_i(\rho_g \bar{u}_g^2/2) - \bar{u}_g(dW_g/dx) \quad (3-2-15)$$

where the first term on the RHS is the interfacial shear stress in the absence of condensation.

### 3-3. Pressure Gradient

A number of general correlations for the pressure gradient in two-phase flow are available<sup>(35-38)</sup>, but these correlations were developed primarily for non-stratified flow in round tubes and fail to take adequately into account the effect of large interfacial mass transfer. Hence, a new correlation was developed for the present case. A momentum balance for a control volume on the steam side, based upon the same assumptions as before, (Fig.2) gives:

$$\frac{d}{dx} (\bar{u}_g W_g) - u_i \frac{dW_g}{dx} = -(H-\delta) \frac{dP}{dx} - (\tau_i + \tau_b) - \rho_g \cdot g \cdot (H-\delta) \sin \theta \quad (3-3-1)$$

which can be rewritten as:

$$\rho_g (H-\delta) \bar{u}_g \frac{d\bar{u}_g}{dx} + (\bar{u}_g - u_i) \frac{dW_g}{dx} = -(H-\delta) \frac{dP}{dx} - (\tau_i + \tau_b) - \rho_g \cdot g \cdot (H-\delta) \sin \theta \quad (3-3-2)$$

Upon employing equation (3-2-15) after replacing  $\bar{u}_g$  with  $(\bar{u}_g - u_i)$  where the first term is now the adiabatic shear stress;

$$\tau_i = \tau_{i,a} + (\bar{u}_g - u_i) \frac{dW_g}{dx} \quad (3-3-3)$$

Substituting equation (3-3-3) into (3-3-2) and rearranging, one obtains an expression for the pressure gradient;

$$\frac{dP}{dx} = -\rho_g \bar{u}_g \frac{d\bar{u}_g}{dx} - \frac{1}{H-\delta} (\tau_{i,a} + \tau_b) - \rho_g \cdot g \cdot \sin \theta \quad (3-3-4)$$

For the momentum balance without mass transfer ( $d\bar{u}_g/dx = 0$ );

$$\left. \frac{dP}{dx} \right|_a = -\frac{1}{H-\delta} (\tau_{i,a} + \tau_b) - \rho_g \cdot g \cdot \sin \theta \quad (3-3-5)$$

By combining equation (3-3-4) and (3-3-5), it is found that;

$$\frac{dP}{dx} = \left. \frac{dP}{dx} \right|_a - \rho_g \bar{u}_g \frac{d\bar{u}_g}{dx} \quad (3-3-6)$$

Thus, the total pressure gradient is represented by the sum of the pressure gradient in the absence of mass transfer and the condensation-induced pressure gradient. The first term on the RHS is negative and the second term is positive in the presence of condensation, so that the total pressure gradient may be positive or negative. Equation (3-3-6) will be checked by choosing a proper equation for calculating the pressure gradient due to the viscous effect.

### 3-4. THICKNESS OF WATER LAYER

Although a considerable literature (39-47) exists on the thickness of falling films in cocurrent gas-liquid flow, or in liquid-only flow, the literature for inclined stratified counter-current flow is much more sparse. Hence, a simplified relation for the film thickness has been developed for the turbulent liquid flow, with or without interfacial shear acting in the opposite-direction of the liquid flow.

#### 3-4-1. Thickness of Water Layer Without Interfacial Shear

The volumetric flow rate per unit width in a falling film can be calculated using a universal velocity distribution:

$$Q = \bar{u} \cdot \delta = \int_0^\delta u \, dy = \nu \int_0^b u^+ \, dy^+ \quad (3-4-1)$$

where

$$u^+ = \frac{u}{u_*} \quad (3-4-2)$$

$$y^+ = \frac{u_* y}{\nu} \quad (3-4-3)$$

The von Karman<sup>(48)</sup> universal distribution is defined:

$$u^+ = y^+ \quad 0 < y^+ < 5 \quad (3-4-4)$$

$$u^+ = -3.05 + 5.0 \ln y^+ \quad 5 < y^+ < 30 \quad (3-4-5)$$

$$u^+ = 5.5 + 2.5 \ln y^+ \quad 30 < y^+ < b \quad (3-4-6)$$

By assuming that these equations, which were developed for fully-developed pipe flow, can be applied to the liquid phase in two-phase flow, it is possible to develop a relationship between the film thickness and the flow rate. One can note that Gazley<sup>(49)</sup> examined the flow of air over a flat plate to the universal basis and demonstrated that the constants and form of the equation for the velocity distribution in the turbulent boundary layer were almost identical to equation (3-4-6). A detailed discussion of this assumption has been given by Dukler, et al<sup>(39)</sup>.

This integration gives

$$\frac{Q}{\nu} = b\{3.0 + 2.5 \ln b\} - 64 \quad (3-4-7)$$

The wall shear stress in the absence of interfacial shear is given by

$$\tau_o = \rho \cdot g \sin \theta \cdot \delta \quad (3-4-8)$$

In the above equations the friction velocity and the dimensionless film thickness are defined by

$$u_* = \sqrt{\tau_o / \rho} = \sqrt{\delta \cdot g \sin \theta} \quad (3-4-9)$$

$$b = \frac{u_* \cdot \delta}{\nu} = \left\{ \frac{g \cdot \sin \theta}{\nu^2} \right\}^{1/2} \delta^{3/2} \quad (3-4-10)$$

Substituting equation (3-4-10) into equation (3-4-7) and rearranging the terms:

$$\delta = \left( \frac{\nu^2}{g \sin \theta} \right)^{1/3} (3.0 + 2.5 \ln \frac{\delta \cdot u_*}{\nu})^{-2/3} \left( 1 + \frac{64}{Re} \right)^{2/3} Re^{2/3} \quad (3-4-11)$$

Using the binomial expansion

$$\left( 1 + \frac{64}{Re} \right)^{2/3} = 1 + \frac{2}{3} \left( \frac{64}{Re} \right) + \frac{\frac{2}{3} \cdot (\frac{2}{3} - 1)}{2} \left( \frac{64}{Re} \right)^2 + \dots \quad (3-4-12)$$

If the minimum value for the turbulent Reynolds number is assumed to be 400, the higher order terms can be neglected since  $(64/Re)^2 \ll 1$ . Therefore:

$$\left( 1 + \frac{64}{Re} \right)^{2/3} = 1 + \frac{2}{3} \left( \frac{64}{Re} \right) \quad (3-4-13)$$

Substituting equation (3-4-13) into equation (3-4-11):

$$\delta = \frac{1 + 42.3 Re^{-1}}{(3.0 + 2.5 \ln \frac{\delta \cdot u_*}{\nu})^{2/3}} \left( \frac{\nu^2}{g \sin \theta} \right)^{1/3} Re^{2/3} \quad (3-4-14)$$

To give the relation between  $\bar{u}$  and  $u_*$ :

$$\tau_o = f \cdot \frac{\rho \bar{u}^2}{2} \quad (3-4-15)$$

By using the Blasius friction formula for the turbulent flow;

$$X = \frac{1 + 42.3 \cdot Re^{-1}}{(3.0 + 2.5 \ln \frac{\delta \cdot u_*}{\nu})^{2/3}} = \frac{1 + 42.3 \cdot Re^{-1}}{(3.0 + 2.5 \ln / 0.028 \cdot Re^{-1/4} + 2.5 \ln Re)^{2/3}} \quad (3-4-16)$$

The parameter, X, is only a function of Reynolds number and shows a small dependence on Reynolds number, as shown in Fig.3:

$$X \approx 0.375 Re^{-0.105} \quad (3-4-17)$$

Substituting equation (3-4-17) into equation (3-4-14):

$$\delta = 0.375 \left( \frac{v^2}{g \sin \theta} \right)^{1/3} Re^{0.562} \quad (3-4-18)$$

This provides a very simple relation between the film thickness and the flow rate, even the film thickness can be obtained using equation (3-4-7) with the information of the wall shear stress. This final relation has the same form as the equations derived by Brauer<sup>(41)</sup> and Brotz<sup>(40)</sup> for free-falling turbulent wall layers, even the constants and the exponents are somewhat different:

$$\delta = 0.435 \left( \frac{v^2}{g} \right)^{1/3} Re^{8/15} \quad (\text{Brauer}) \quad (3-4-19)$$

$$= 0.172 \left( \frac{v^2}{g} \right)^{1/3} Re^{2/3} \quad (\text{Brotz}) \quad (3-4-20)$$

### 3-4-2 Thickness of Water Layer With Interfacial Shear

From a force balance for the control volume in a liquid flow system shown in Fig. 4, one obtains

$$-\tau_1 \Delta z - \tau \Delta z + \rho \cdot g \cdot \sin \theta (\delta - y) \cdot \Delta z - \Delta p (\delta - y) = 0 \quad (3-4-21)$$

which gives:

$$\tau = (\rho \cdot g \cdot \sin \theta - \frac{\Delta p}{\Delta z}) (\delta - y) - \tau_1 \quad (3-4-22)$$

Assuming  $\rho \cdot g \cdot \sin \theta \gg \Delta p / \Delta z$ , the equation for the distribution of shear stress can be obtained:

$$\tau = \rho \cdot g \cdot \sin \theta (\delta - y) - \tau_1 \quad (3-4-23)$$

From the above equation,

$$\tau_0 = \rho g \sin \theta \delta - \tau_1 \quad (3-4-24)$$

The shear stress is related to the velocity gradient by:

$$\tau = \rho (v + v_E) \frac{du}{dy} \quad (3-4-25)$$

The relation between the shear stress and the velocity gradient is defined using the Prandtl mixing length theory and neglecting the kinematic viscosity:

$$\tau = \rho \kappa^2 y^2 \left| \frac{du}{dy} \right| \frac{du}{dy} \quad (3-4-26)$$

Where  $\kappa$  is the von Karman constant ( $=0.4$ ).

Combining equations (3-4-23) and (3-4-26)

$$\rho \kappa^2 y^2 \left| \frac{du}{dy} \right| \frac{du}{dy} = \rho \cdot g \cdot \sin \theta (\delta - y) - \tau_1 \quad (3-4-27)$$

When  $y_2 = \delta - \tau_1 / \rho \cdot g \cdot \sin \theta$ , the shear stress is zero and the slope of the velocity distribution is reversed.

The velocity distribution is solved separately in three regimes. In the region of viscous sublayer, where  $0 < y < y_1$ ,

$$\rho \cdot g \cdot \sin \theta (\delta - y) - \tau_1 = \mu \frac{du}{dy} \quad (3-4-28)$$

By integrating, the velocity at  $y_1$  can be obtained:

$$u_1 = \frac{1}{\mu} \{ (\rho \cdot g \cdot \sin \theta \cdot \delta - \tau_1) y_1 - \rho \cdot g \cdot \sin \theta \frac{y_1^2}{2} \} \quad (3-4-29)$$

The thickness of the viscous sublayer,  $y_1$ , is obtained from

$$y_1 = C \frac{v}{u_*} = 11.6 \frac{v}{\sqrt{\tau_0 / \rho}} \quad (3-4-30)$$

where the value of 11.6 has been taken for the value of the constant,  $C^{(43)}$ .

In the region ( $y_1 < y < y_2$ ) where the shear stress is positive, Eq.(3-4-27) is solved by substituting

$$\rho \kappa^2 y^2 \left( \frac{du}{dy} \right)^2 = \rho g \sin \theta (\delta - y) - \tau_1 \quad (3-4-31)$$

Taking the square root, integrating once and inserting the boundary condition:  $u = u_1$  at  $y = y_1$ ;

$$u = u_1 + \frac{\sqrt{\tau_0 / \rho}}{\kappa} \left\{ 2(\sqrt{1 - \beta \cdot y'} - \sqrt{1 - \beta \cdot y_1'}) + \ln \frac{\sqrt{1 - \beta \cdot y'} - 1}{\sqrt{1 - \beta \cdot y_1'} - 1} + \ln \frac{\sqrt{1 - \beta \cdot y_1'} + 1}{\sqrt{1 - \beta \cdot y'} + 1} \right\} \quad (3-4-32)$$

$$\text{where } y' = y / \delta \quad (3-4-33)$$

$$\beta = \frac{\rho \cdot g \cdot \sin \theta \cdot \delta}{\tau_0} \quad (3-4-34)$$

In the region ( $y_2 < y < \delta$ ) where the shear stress is negative, Eq.(3-4-27) is solved by substituting

$$\rho \kappa^2 (\delta - y) \left( \frac{du}{dy} \right)^2 = - \{ \rho \cdot g \cdot \sin \theta (\delta - y) - \tau_1 \} \quad (3-4-35)$$

This yields

$$u = u_2 + \frac{\sqrt{\tau_0 / \rho}}{\kappa} \left\{ 2(\sqrt{-1 + \beta \cdot y'} - \sqrt{-1 + \beta \cdot y_2'}) + \sqrt{-1 + \beta} \left( \ln \frac{\sqrt{-1 + \beta \cdot y'} - \sqrt{-1 + \beta}}{\sqrt{-1 + \beta \cdot y_2'} - \sqrt{-1 + \beta}} + \ln \frac{\sqrt{-1 + \beta \cdot y_2'} + \sqrt{-1 + \beta}}{\sqrt{-1 + \beta \cdot y'} + \sqrt{-1 + \beta}} \right) \right\} \quad (3-4-36)$$

The quantity of liquid flowing through the film can be calculated from

$$Q = \int_0^{\delta} u \, dy = \int_0^{y_1} u \, dy + \int_{y_1}^{y_2} u \, dy + \int_{y_2}^{\delta} u \, dy \quad (3-4-37)$$

These relationships allow calculation of the liquid film thickness from a knowledge of the liquid flow rate and the interfacial shear stress. The velocity distributions obtained from equations (3-4-29), (3-4-32) and (3-4-36) are shown in Fig. 5, as functions of the dimensionless interfacial shear stress  $\gamma = \tau_i / \rho \cdot g \cdot \sin \theta \cdot \delta$ . As expected, the interface velocity becomes smaller as the interfacial shear stress increases. The relation between the dimensionless film thickness and the Reynolds number is shown in Fig. 6. As expected, the film thickness increases as the interfacial shear stress increases, in contrast to cocurrent flow.

### 3-5. Turbulence-centered Model

It is of interest to attempt to develop a model for steady-state, turbulent, stratified flows with condensation, which may be expected to exhibit similarities to the corresponding flows without condensation (or with small mass transfer, such as absorption of a slightly soluble gas).

Turbulent gas absorption models, which relate the mass transfer coefficient to the local turbulent properties, have been obtained by two quite different approaches. Fortescue, et al.<sup>(50)</sup> suggested that the liquid is brought to the surface in the form of large eddies. The length scale of the eddies is taken to be the integral scale of the turbulence, and the maximum velocity in the eddy is assumed to be proportional to the turbulent intensity,  $u_t$ . Solution of the transport equation with periodic boundary conditions leads to:

$$k_L = 1.46 \left( \frac{D_L \cdot u_t}{\lambda_t} \right)^{1/2} \quad (3-5-1)$$

They compared this result with that predicted by the Higbie<sup>(51)</sup> penetration theory, where  $t_{exp}$  is taken to be  $\lambda_t / u_t$ , which gives:

$$k_L = 1.12 \left( \frac{D_L \cdot u_t}{\lambda_t} \right)^{1/2} \quad (3-5-2)$$

This gives a value roughly 20% lower.

Banerjee, et al.<sup>(52)</sup> and Lamont, et al.<sup>(53)</sup>, on the other hand, considered that small eddies may dominate the transport process at high Reynolds numbers. Thus, the mass transfer rate at the free interface can be related to the rate of viscous dissipation in the turbulent flow near the surface. It is, therefore, a reasonable hypothesis to consider the mass transfer rate as being governed by diffusion into the smallest eddies. The diffusion equation was solved by assuming a self-preserving concentration profile in the boundary layer. Upon integrating with respect to wave number, using the energy spectrum for fully-developed turbulence by Kovaszny<sup>(54)</sup>, the result was

$$k_L \sim \left( \frac{D_L^2 \cdot \epsilon}{\nu} \right)^{1/4} \quad (3-5-3)$$

The two models (large and small eddy model) are thus similar, but differ in the size of eddies considered to be controlling. Fortescue, et al. chose the large eddies because they contain most of the turbulent energy. The view of Lamont, et al. is that the small-scale motions may be more efficient for interface transfer, in spite of their low energy, because they are responsible for microscopic mixing at high turbulence levels. Brumfield, et al.<sup>(18,21)</sup> examined several sets of mass transfer data and presented a synthesis of the large-eddy model (energy-containing motion) and the small-eddy model (energy dissipating motion):

$$k_L = 0.25 \sqrt{D_L} \left( \frac{u_t^3}{\nu \cdot \lambda_t} \right)^{1/4} \quad \text{Re}_t > 500 \quad (3-5-4)$$

$$= 0.7 F(\alpha_e) \left( \frac{D_L u_t}{\lambda_t} \right)^{1/2} \quad \text{Re}_t < 500 \quad (3-5-5)$$

By using a Reynolds analogy, it is readily found that the analogous dimensionless heat transfer equations are<sup>(19,20)</sup>

$$\text{Nu}_t = 0.25 \text{Re}_t^{3/4} \text{Pr}^{1/2} \quad \text{Re}_t > 500 \quad (3-5-6)$$

$$= 0.7 F(\alpha_e) \text{Re}_t^{1/2} \text{Pr}^{1/2} \quad \text{Re}_t < 500 \quad (3-5-7)$$

By considering the strong effect of the sheared and disturbed interface on the turbulent properties, the turbulent velocity and length scale may be properly defined to apply this model to the steam condensation.

### 3-6. Interfacial Wave Structure

In two phase flow, the phase interface is generally characterized by a complex wave structure due to the highly agitated steam-water interactions. The nature and characteristics of interface waves are exceedingly complex. It has been repeatedly mentioned in the literature that values of liquid phase heat or mass transfer coefficients obtained experimentally considerably exceed the values calculated on the basis of the various theoretical models. The presence of the waves is known to modify the turbulence structure in the gas phase. These waves must also modify the flow pattern in the liquid in order to account for the increase in mass and heat transfer<sup>(55,56)</sup>, which cannot be accounted for by interfacial area increase alone due to the wave.

Since the instantaneous film thickness is a randomly fluctuating quantity, time-series analysis is generally used to provide statistical parameters, such as the mean film thickness, central moments, probability density function, spectral density and covariance function. In this research, a method for the direct measurement of probability distribution of liquid as a function of film height was adopted, using needle contact probes.

The probability of liquid phase at a particular probe location is defined as the fraction of time that the probe spends in contact with the liquid during some time interval:

$$P(S_0) \triangleq \frac{1}{t_0} \int_t^{t+t_0} \langle S(t') - S_0 \rangle dt' \quad (3-6-1)$$

$$\langle S(t) - S_0 \rangle = \begin{cases} 1: S(t) > S_0 & \text{if the probe is in liquid} \\ 0: S(t) > S_0 & \text{if the probe is in gas} \end{cases}$$

The integration time,  $t_0$ , is chosen to be long enough to yield a statistically stable probability. The mean water layer thickness is defined as:

$$\delta = \int_0^1 S(P) dP \quad (3-6-2)$$

The mean wave amplitude is defined as:

$$S_1 = \overline{|S - \delta|} = \int_0^1 |S(P) - \delta| dP \quad (3-6-3)$$

The square root of the variance,  $S_2$ , is defined as the intensity of film thickness fluctuation and may be expressed by:

$$S_2 = \overline{[(S - \delta)^2]}^{1/2} = \left[ \int_0^1 [S(P) - \delta]^2 dP \right]^{1/2} \quad (3-6-4)$$

Instead of obtaining spectral densities by time-series analysis, the number of zero crossings per second was measured at a particular position of the probe. The distribution of the frequency of crossings as a function of probe elevation is integrated with respect to the probability distribution of the water layer thickness to give an average wave frequency

$$\bar{N}_f = \int_0^1 N_f(P) dP \quad (3-6-5)$$

## 4. EXPERIMENTAL PROCEDURE

### 4-1. Test Facility

The steam-water countercurrent flow contactor is made up of three sections, consisting of an upper and a lower plenum, together with a test section (Fig.7). The upper and the lower plena are designed to insure smooth exit and entry of steam and water. The test section is provided with the appropriate instrumentation to determine the local condensation rates, heat transfer coefficients and interfacial wave structure. The test section is completely insulated, except for pyrex windows fitted into both side plates. Liquid entry is through a porous (100 $\mu$ ) stainless steel plate 2.4mm thick. For water flow rates of 0.3 ~ 3.0 kg/s, the pressure drop through the porous medium is 0.25 ~ 2.8 psi, which is adequate to establish a smooth gravity-fed flow at the liquid entrance of the test section. The steam entrance region in the lower plenum contains a set of screens and a perforated plate to establish uniform flow. In addition, honeycomb material is inserted to act as turning vanes through the 90° bend. The water leaving the test section is drained by gravity through a set of fine mesh screens. The residual water which has not been thus removed is trapped in a small water receptacle formed by the side wall and a flow divider, and is drained into a holding tank. The water level in the holding tank is controlled by a magnetic level controller and a solenoid valve. The water circulates through the heat exchanger into the water entrance plenum. The inlet water temperature can be controlled to any desired temperature by means of a heat exchanger. Steam from the building supply is throttled down to a slightly superheated temperature at atmospheric pressure after removing any condensate in the supply line by means of a steam separator. This arrangement made it possible to have dry, slightly superheated steam join smoothly in parallel flow with the water layer at the bottom of the test section. The overall system of the test facility is shown in Fig. 8.

### 4-2. MEASUREMENT SYSTEM

#### 4-2-1. Measurement of flow properties

The steam flow rates were measured at six stations: 0.26m, 0.44m, 0.62m, 0.8m, 0.98m, and 1.27m from the steam entrance along the channel center (for the experiment of aspect ratio 10, the stations are 0.22m, 0.4m, 0.58m, 0.76m, 0.93m, and 1.27m). This was accomplished by integrating the measured steam velocity profile at each station, assuming two-dimensional flow. The two-dimensionality assumption was verified by pitot tube traverses across the width of the test section, and the integration method was checked by measuring the steam velocity profiles across the

height of the test section with zero water flow. Upon comparing the measured inlet steam flow rate with the calculated local steam flow rate, it was found that the deviation was less than  $\pm 2\%$ , as shown in Fig.9. The pitot tubes, which were electrically heated to prevent steam from condensing inside the tubes, were attached to a common traversing table, allowing them to be positioned at selected elevations in the test section within 0.05mm. The difference between total and static pressure for each pitot tube was measured by means of a diaphragm-type differential pressure transducer. The inlet steam flow rate was calculated from measurements of the differential pressure, absolute pressure and the thermodynamic state of the incoming steam, using a 50mm inlet steam venturi. The pressure difference and the absolute pressure were measured with diaphragm-type differential pressure transducers. The static pressures at the same locations as the pitot tubes in the test section were measured by wall-mounted pressure taps. Inlet and outlet temperatures of the steam and water were measured by using K-type (Chromel-Alumel) thermocouples connected to a zero-point reference junction. The water inlet flow rate was determined by measuring the pressure drop across a 32mm water venturi meter.

#### 4-2-2. Measurement of interfacial wave structure

A large variety of techniques has been developed for the determination of the interfacial wave structure; good reviews are given by Collier, et al.<sup>(57)</sup> and Hewitt<sup>(58)</sup>. Needle contact probes have been adopted for the present research because of the relatively thicker films and also the presence of condensation.

These probes tend to under-estimate the void fraction in bubbly flows because of deflection of the bubbles before and upon hitting the probe, in view of their low effective mass. However, this problem appears to be minimal in measuring interfacial waves with small-diameter probes. Another serious difficulty in the application of this type of probe is the uncertainty about contact hysteresis. It may be possible for the probe to remain in extended contact with the liquid by dragging out a filament of liquid which has to break before a break in contact is recorded. This problem has never been thoroughly resolved, although many workers consider it unlikely to be of significance. In addition to this problem, polarization effect and electrochemical attack of the probe should be avoided.

To minimize these effects, the probe, consisting of a 0.5mm stainless steel needle with a sharpened tip, was made non-wetting by application of a teflon coating, except at the very tip. A.C. excitation at 10 KHz is used to avoid the polarization problem (Fig.10). To obtain an accurate position of the probe tip relative to the bottom wall, the probe is traversed by means of a micrometer attachment, first to the wall, and then returned to the interface. In practice, the raw probe output was quite close to a series of step functions, so that the thresholding voltage could be easily determined to give an accurate local void

fraction. Using these probes, the distribution of probability and number of crossings for liquid contact with the probe as a function of probe elevation could be measured. The probability is equivalent to the time-average liquid volume fraction,  $1-\alpha$ , as a function of height. The probe signals were fed directly into a PDP-11/34 computer through a A/D converter with, usually, a sampling time and sampling frequency of about 5 seconds and 2000Hz. The signal obtained through A.C. circuitry by the teflon-coated probe at gas-liquid interface is shown in Fig. 11-a, and compared with the signal from the contact of the probe with the solid metal, as shown in Fig. 11-b.

#### 4-2-3. Data Acquisition System

The output signals were sampled and processed by a PDP-11/34 minicomputer which is equipped with a 16 channel AR-11 A/D converter with a usual sampling time and sampling frequency of 10 seconds and 240Hz ( 5 seconds and 2000Hz for needle probe). The reduced data were transferred to magnetic discs for permanent storage and displayed on the CRT terminal located in the laboratory.

## 5. DATA PRESENTATION AND ANALYSIS

### 5-1. Typical Data

The data reported here are for the following inclination angles and aspect ratios:

- i)  $\theta$ , inclination angle, = 4 and B/H, aspect ratio, = 5(381 mm - 75 mm)
- ii)  $\theta$  = 30 and B/H = 5(381 mm - 75 mm)
- iii)  $\theta$  = 33 and B/H = 10(381 mm - 38 mm)

Some of the data for  $\theta$  = 33 and B/H = 10 were given in a previous report<sup>(59)</sup>. A test matrix showing the conditions for the measured water and steam flow rates and inlet water temperatures for the latter two cases is shown in Figures 12-13. The range of test conditions was restricted for two reasons: i) high water and/or steam flow rates produce surface instabilities, leading to bridging and/or flooding; ii) with cold water and low steam flow rates, complete condensation occurs within the test section, which results in large pressure pulses (water hammer). The reproducibility of the steam flow measurements was checked by duplicate measurements at each station on different days, resulting in agreement within  $\pm 2\%$ . The presence of a uniform velocity profile at the steam entrance to the test section was verified by the pitot tube measurements, as shown in Fig. 14, and the development of the velocity profile in the test section with zero water flow is shown in Fig. 15.

Typical steam flow rates as a function of axial position are shown in Figures 16-17, together with curves obtained by fitting a fourth-order polynomial to the data. As expected, the condensation rate increases with increasing steam inlet flow rate, increasing water flow rate and decreasing water inlet temperature.

### 5-2. Heat Transfer Coefficients

Local heat transfer coefficients were calculated from equations (3-1-4) and (3-1-7).  $dW/dz$  was calculated by differentiating the fourth-order polynomial fitted through the measured steam flow rate data. Typical local heat transfer data (Fig. 18) show the expected trends. The local heat transfer coefficient decreases smoothly from  $x = 0$  (steam inlet, water exit), where the shear stress and wave amplitudes are maximum, towards a minimum at  $x = L$ . Increasing the steam flow rate increases the heat transfer coefficient.

In the tests reported here,  $h$  varied from about 1-13  $\text{kw/m}^2\text{K}$  for an inclination angle  $\theta$  = 4, and an aspect ratio B/H = 5, 5-18  $\text{kw/m}^2$  for  $\theta$  = 30 and B/H = 5 and 5-25  $\text{kw/m}^2\text{K}$  for  $\theta$  = 33 and B/H = 10. The effect of increased inclination angle and aspect ratio is an increase in heat transfer coefficient. In a comparable study, Segev, et al.<sup>(16)</sup> obtained local heat transfer coefficients ranging from 8 to 28  $\text{kw/m}^2\text{K}$ . On the other hand, the data of Cook, et al.<sup>(60)</sup> were higher (5-45  $\text{kw/m}^2\text{K}$ ). This is to be expected, since the test section was nearly vertical and the

aspect ratio was 10. The local Nusselt number can be considered to be principally a function of the local gas and liquid Reynolds numbers, and possibly the liquid Prandtl number, since the thermal resistance lies principally on the liquid side. An empirical power-law correlation would then be of the form.

$$Nu = n_1 Re_g^{n_2} Re_l^{n_3} Pr^{n_4} \quad (5-2-1)$$

For  $\theta = 33^\circ$  and  $B/H = 10$ , described in a previous report<sup>(59)</sup>, the data were separated according to wave structure and correlated by least-square fitting, giving the relations:

$$Nu = 0.173 Re_g^{0.027} Re_l^{0.49} Pr^{0.42} \quad \text{(smooth interface region)} \quad (5-2-2)$$

$$Nu = 3.43 \times 10^{-10} Re_g^{2.1} Re_l^{0.56} Pr^{1.16} \quad \text{(rough interface region)} \quad (5-2-3)$$

Thus,  $Re_g$  has practically no effect on the Nusselt number in the smooth interface region, but dominates in the rough region. The range of variation of the Prandtl number was at most a factor of three, so that no significance is attached to the variation in Prandtl number exponent in passing from a smooth to a rough interface.

For  $\theta = 4^\circ$  and  $B/H = 5$ , the data were correlated by least-square fitting, as shown in Fig. 19, giving:

$$Nu = 0.63 \times 10^{-5} Re_g^{0.9} Re_l^{0.75} Pr^{0.81} \quad (5-2-4)$$

For the inclination angle  $30^\circ$  and the aspect ratio 5, as shown in Fig. 20:

$$Nu = 0.135 \times 10^{-3} Re_g^{0.35} Re_l^{1.0} Pr^{0.56} \quad (5-2-5)$$

It is seen that there are large variations in the Reynolds number exponents for these different cases. From the visual observations and the measurements of the interfacial wave structure, a variety of wave structures were observed, depending on the inclination angle and aspect ratio. These may be attributed, in large part, to the differing aspect ratios and inclination angles and, very importantly, the differing steam-water relative velocities at the same values of the gas and liquid Reynolds numbers. The Froude number based on relative velocities may therefore be a better correlating variable. A similar discrepancy has been found in the comparable study of the correlations by Segev, et al.<sup>(16)</sup>, who classified their condensation data into several groups, corresponding to smooth interface, rough interface and bypass region (flooding). A detailed comparison between Segev, et al. correlation and the present correlation for the inclination angle  $33^\circ$  and aspect ratio 10 has been presented previously<sup>(61)</sup>.

By carefully considering the effect of wave structure, a general correlation for Nusselt number may be obtained, which is not explicitly dependent on inclination angle or aspect ratio. This will be investigated in the future.

### 5-3. Interfacial Shear Stress

Equations (3-2-1), (3-2-2) and (3-2-3) can be used to calculate the interfacial shear stress. For this turbulent liquid flow with wavy interface, the interface velocity is assumed to be the average liquid velocity. This assumption is in agreement with results of Akai, et al.<sup>(30)</sup> for cocurrent turbulent flow. The friction factor,  $f_1$ , in the absence of condensation was calculated from the correlation (3-2-5) and (3-2-6), despite the fact that they were obtained for cocurrent flow, since the adiabatic shear stress was much smaller than the condensation shear stress in the present steam-water flows. The data for the interfacial shear stress using equation (3-2-1) are compared with this model (Eqn. 3-3-3) (Figures 21-22) for the aspect ratio 5 and inclination angles  $40^\circ$  and  $30^\circ$ . The results are in good agreement with the values predicted by this modified Linehan model, substantiating the assumptions.

### 5-4. Pressure Gradient

Typical pressure distributions as a function of axial position are shown in Fig. 23, together with curves obtained by fitting a fourth-order polynomial to the data. As described in Section 3-3, the total pressure gradient in the presence of mass transfer can be represented by the summation of the pressure gradient due to viscous and gravity effects and the condensation-induced pressure gradient:

$$\frac{dP_g}{dx} = \frac{dP_g}{dx} \Big|_a - \rho_g \bar{u}_g \frac{d\bar{u}_g}{dx} \quad (3-3-6)$$

The first term in the RHS, the pressure gradient due to viscous and gravity effect, is always negative, whereas the second term in the RHS, the condensation-induced pressure gradient, is positive in the presence of mass transfer. Therefore, the pressure gradient is dependent on the condensation rate and flow properties in the presence of condensation. This is shown in Fig. 23, which indicates that the pressure gradient increases with increasing condensation rates.

In the experiments with  $\theta = 30^\circ$ , a negative pressure gradient sometimes occurs near the steam exit, where the condensation effect is small. The measured pressure gradients are compared with values calculated by (3-3-5) in the absence of condensation, together with (3-2-5, 3-2-6) for the interfacial shear stress and (3-2-3) for the wall shear stress. Good agreement is shown, as in Fig. 24. Equation (3-3-6) for the pressure gradient in the

presence of condensation is compared with the measured values for  $B/H = 5$  and  $\theta = 4$  and  $30$ . As shown in Figures 25-26, the calculated values show good agreement with the measured values for the pressure gradient. Hence this simple model is promising for the evaluation of pressure gradient in stratified countercurrent flow in the presence of condensation.

#### 5-5. Thickness of Water Layer

In section (3-4), an equation for the water layer thickness has been derived, using Prandtl mixing length theory and the turbulent transport equation for the relation between the shear stress and the velocity gradient, and assuming a linear distribution of shear stress in the water layer, either with or without the interfacial shear stress. The derived equation (3-4-18) for no interfacial shear stress is compared with the Brauer and Brotz equations (3-4-19, 3-4-20) for free-falling turbulent wall layers, and the Nusselt equation for laminar film flow, by non-dimensionalizing the thickness:

$$\delta' = \delta / \left( \frac{v^2}{g \cdot \sin \theta} \right)^{1/3} \quad (5-5-1)$$

Equations (3-4-18, 3-4-19, 3-4-20) and the Nusselt equation are non-dimensionalized as follow:

$$\delta' = 0.375 \quad Re^{0.562} \quad (\text{present analysis}) \quad (5-5-2)$$

$$\delta' = 0.435 \quad Re^{8/15} \quad (\text{Brauer}) \quad (5-5-3)$$

$$\delta' = 0.172 \quad Re^{2/3} \quad (\text{Brotz}) \quad (5-5-4)$$

$$\delta' = 1.44 \quad Re^{1/3} \quad (\text{Nusselt}) \quad (5-5-5)$$

These relationships are compared with experimental data for the various inclination angles in the absence of the interfacial shear stress in Fig. 27. The experimental data fit very well with the present analysis, even though somewhat underestimated at high Reynolds numbers. At high Reynolds numbers, the interface is characterized by waves with finite amplitude. The measured thickness is taken to be at the point where  $\alpha = 0.5$ . Since the equation for the thickness was derived neglecting the interface wave structure, it is questionable whether the point where  $\alpha = 0.5$ , which corresponds to the local time-mean film thickness, would be the same as the local volume-average film thickness. Another point is that large lumps of liquid, which carry a significant portion of all the liquid flowing, move down the interface with no change in interface shape. Hence, the smooth film velocity distribution analysis will predict primarily the thickness of the wave troughs, rather than the mean film thickness. This will be recognized later in the comparison with the data in the presence of interfacial shear stress.

Empirical relations between the dimensionless thickness and the Reynolds number in the presence of interfacial shear stress were obtained by least-square fitting. The preliminary results for each inclination angle and aspect ratio are summarized as follows (Fig. 28) :

For  $\theta = 4$  and  $B/H = 5$ ,

$$\delta' = 0.471 \text{ Re}^{0.58} \quad (5-5-6)$$

For  $\theta = 30$  and  $B/H = 5$ ,

$$\delta' = 0.205 \text{ Re}^{0.66} \quad (5-5-7)$$

For  $\theta = 33$  and  $B/H = 10$ ,

$$\delta' = 0.388 \text{ Re}^{0.66} \quad (5-5-8)$$

More data will be required in order to obtain a general correlation for the dimensionless film thickness as a function of inclination, aspect ratio and liquid Reynolds number.

The water layer thicknesses were calculated, using the method described in section 3-4-2, according to the interfacial shear stress and compared with the measured mean water layer thicknesses, as shown in Fig.29. As recognized above in the case of zero interfacial shear stress, the calculated thicknesses are smaller than the measured mean thickness. The underestimation of the thickness may be attributed to the presence of the interface wave, as described above. This is verified in the comparison of the calculated thickness with the measured distance between the bottom wall and the trough of the wave. As shown in Fig.30, the calculated thicknesses agree well with the distance to the trough.

#### 5-6. Turbulence-centered Model

It is interesting to see whether a turbulence-centered model for steam condensation may be developed, which is similar to those which have been presented for absorption of slightly soluble gases at smooth interfaces of cocurrent flows of gases and turbulent liquids. In terms of the analogous dimensionless heat transfer equations, the turbulence equation is (Section 3-5):

$$\text{Nu}_t = 0.25 \text{ Re}_t^{3/4} \text{ Pr}^{1/2} \quad \text{Re}_t > 500 \quad (3-5-6)$$

$$= 0.7 F(\alpha_e) \text{ Re}_t^{1/2} \text{ Pr}^{1/2} \quad \text{Re}_t < 500 \quad (3-5-7)$$

Evaluation of the model depends on the accuracy with which the turbulent velocity and length scales,  $u_t$  and  $\lambda_t$ , can be predicted. For horizontal cocurrent steam water flow, the scales

used in the gas absorption model,  $u_t = 0.3 \bar{u}_\ell$  and  $\lambda_t = \delta$ , result in a good fit of the measured  $h$  and the value predicted by turbulence equation (3-5-6)<sup>(20)</sup>. However, for countercurrent steam-water flow in an inclined channel, the comparison between measured and predicted values reveals large discrepancies<sup>(59)</sup>. Appropriate values are difficult to define for the turbulent velocity scale,  $u_t$ , and length scale,  $\lambda_t$ , due to the highly agitated steam-water interface and large condensation rates, which affect the interfacial shear stress. This is not considered explicitly in the turbulence equation.

Considering the strong effect of the sheared and disturbed interface on the turbulence properties, the turbulent velocity and length scales may be defined in terms of interfacial parameters as follows:

$$u_t = u_* = \sqrt{\tau_i / \rho} \quad (5-6-1)$$

$$\lambda_t = \delta \quad (5-6-2)$$

Improved correlations, using these interfacial scales, are shown in Figures 31-33, depending on the inclination angle and aspect ratio, and are expressed as follows:

For the inclination angle  $40^\circ$  and aspect ratio 5:

$$Nu_t = 0.134 Re_t^{0.96} Pr^{0.5} \quad (5-6-3)$$

For the inclination angle  $30^\circ$  and aspect ratio 5:

$$Nu_t = 0.09 Re_t^{1.05} Pr^{0.5} \quad (5-6-4)$$

For the inclination angle  $33^\circ$  and aspect ratio 10:

$$Nu_t = 0.053 Re_t^{1.13} Pr^{0.5} \quad (5-6-5)$$

Those correlations are compared in Fig. 34 with the turbulence equation (3-5-6) and the Jensen correlation<sup>(62)</sup> for cocurrent steam-water flow:

$$Nu_t = 0.14 Re_t^{1.0} Pr^{0.5} \quad (5-6-6)$$

The dependence upon the turbulent Reynolds number has been increased compared to the smooth-interface eddy-cell models. This is reasonable, however, since the influence of the gas Reynolds number, which is crucial for the rough interface, appears only in the length and velocity scales. This implies, in turn, a larger Reynolds number exponent.

Linehan<sup>(11)</sup> suggested that the Stanton number for condensation of steam at the surface of cocurrent-flow water layer was a constant, based upon the hypothesis of constant eddy diffusivity. Jensen<sup>(62)</sup> argued that the appropriate constant grouping was  $St \cdot Pr^{0.5}$ , based upon an examination of some gas

absorption data, together with his own cocurrent steam-water data. In either case, the Nusselt number is then proportional to the Reynolds number. As noted above, the range of Prandtl number in the present steam-water data is too narrow to allow meaningful determination of the Prandtl number exponent, but the Reynolds number exponent is greater than one, as expected. This is because the influence of the steam flow on the interfacial transport and wave structure is greater in countercurrent flow at an inclination of  $30^\circ$  and  $33^\circ$  than in horizontal cocurrent flow. The increase in Reynolds number exponent for the Linehan model over that for the model of Brumfield, et al.<sup>(18)</sup> can be attributed to the dominant effect of the interfacial condensation on the interfacial shear stress, and hence on the Stanton number. The Brumfield, et al. equation was based entirely on data for the absorption of slightly soluble gases, where such an effect would be negligible. Note that the very weak (or zero) dependence of the Stanton number on the Reynolds number implies that the choice of turbulent length scale has little effect. On the other hand, the frictional characteristics come into play in the choice of the velocity scale.

#### 5-7. Interfacial Wave Structure

The interfacial wave structure was obtained by traversing the micrometer-mounted needle contact probe coated with teflon, using definitions given in Section 3-6. The reproducibility of the interfacial wave structure measurements was checked by duplicate measurements at each station on different days, resulting in good agreement in each probability distribution and in the interfacial parameters calculated from the distributions, as shown in Fig. 35. Typical data for probability distributions as a function of probe heights are shown in Fig. 36. After getting the distribution of the frequency of crossings as a function of probe height, the distribution of the frequency of crossings can be depicted as a function of probability, as shown in Fig. 37.

The interfacial parameters, such as mean water layer thickness, mean wave amplitude, r.m.s. wave amplitude and average frequency of crossings, can be calculated from these distributions. The slopes in the mean water layer thickness have wide variations, depending on inlet flow conditions. In general, the mean water layer thickness has a increasing slope along the direction of flow with increasing inlet water and steam flow rates, as shown in Fig. 38 with typical data, which exhibits characteristics similar to the condensation rates and interfacial shear stress. The increase in condensation rates with increasing steam and water flow rates results in increase in the interfacial shear stress, which is attributed to the increase in water layer thickness, as described in Section 5-5. As the inlet water temperature increases, the slope of the water layer thickness becomes flat. In particular, at almost saturated inlet water temperatures where the condensation rates can be negligible, the water layer thickness is almost the same through the whole channel. This is attributed to the uniform distribution of local

flow properties, depending on the inlet flow conditions in the absence of heat and mass transfer. At these conditions, it appears from the data that the interfacial parameters, such as the mean and r.m.s. wave amplitude and average frequency of crossings, also have uniform distributions along the channel. This result suggests reasonable verification for using local flow properties in finding turbulent correlations. Additionally, the mean water layer thickness, calculated from equation (3-6-2) is shown to be in good agreement with the thickness for 50% probability, as shown in Fig. 39. This is a great convenience in finding the mean water layer thickness without measuring the probability distribution as a function of height, which is very laborious.

The r.m.s. wave amplitude, in general, increases with increasing inlet water temperature and steam flow rate due to the combination of the sheared and disturbed interface. Even the overall magnitudes are shifted with increasing inlet water temperature, the r.m.s. wave amplitude has a uniform distribution along the channel as the water layer thickness, especially at saturated inlet water temperature. The increase in r.m.s. wave amplitude with increasing steam flow rate suggests that the interfacial shear stress plays an important role in the growth of the wave, which is consistent with the result of Telles and Dukler<sup>(26)</sup> for the case of air-water horizontal flow. The mean wave amplitude, defined by equation (3-6-3), may have a certain functional dependence on the r.m.s. wave amplitude, as shown in Figure 40, since the mean wave amplitude depends on the local flow properties in a similar way as the r.m.s. wave amplitude. The detailed analysis for the interfacial wave structure will be demonstrated in the next report.

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Fig. 1 Control Volume for Energy Balance

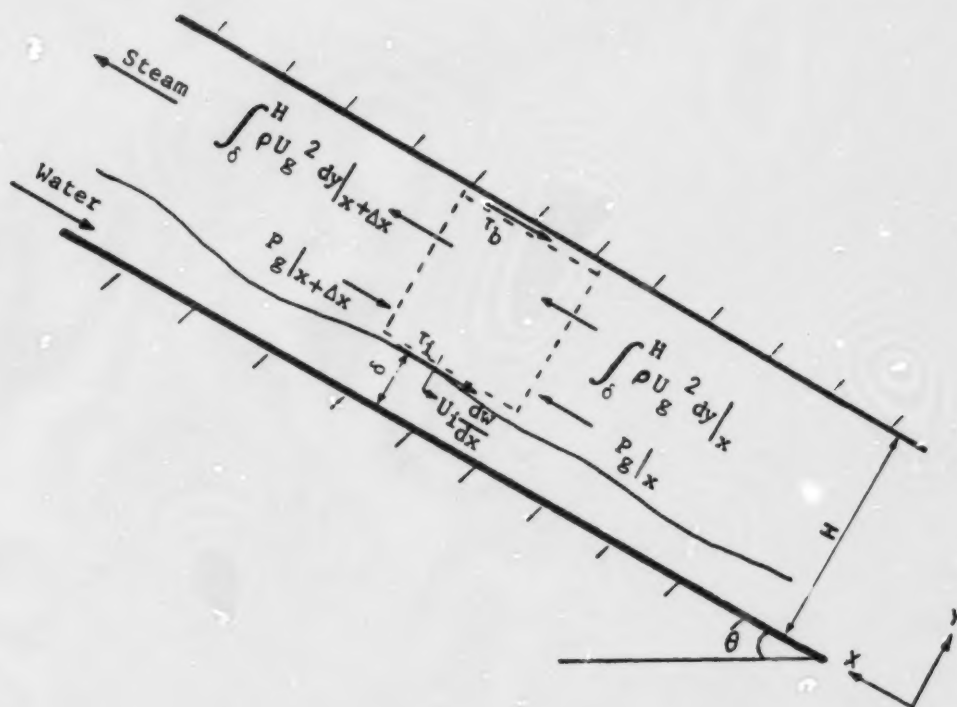


Fig. 2 Control Volume for Momentum Balance at Steam Side

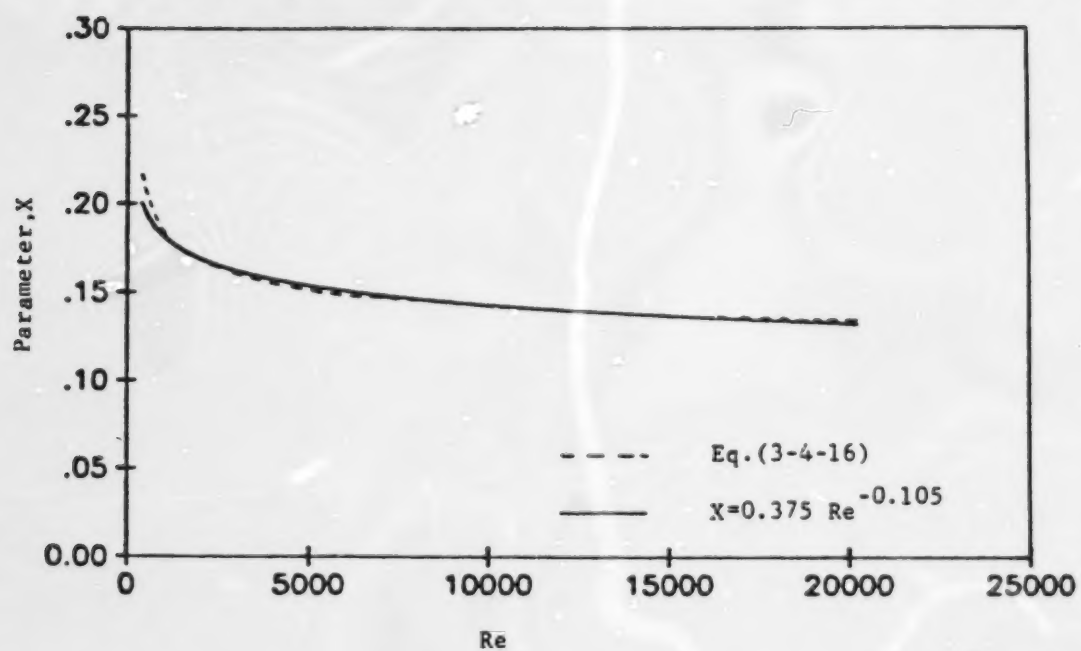
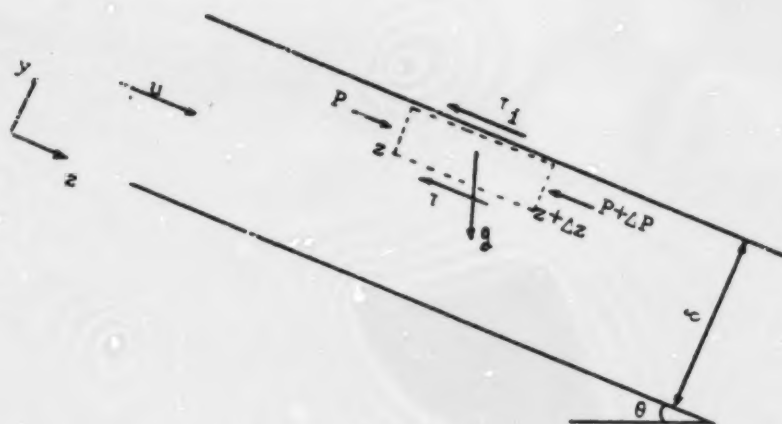
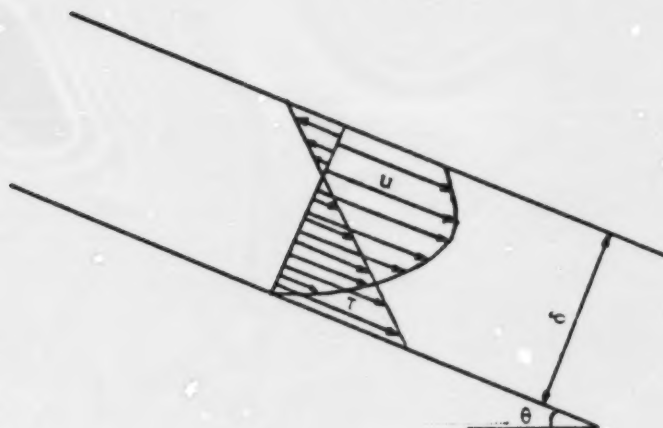


Fig. 3 Dependence of parameter X on Reynolds number



a) Liquid flow element



b) Expected velocity & shear stress distribution

Fig. 4 Assumed flow characteristics in water layer

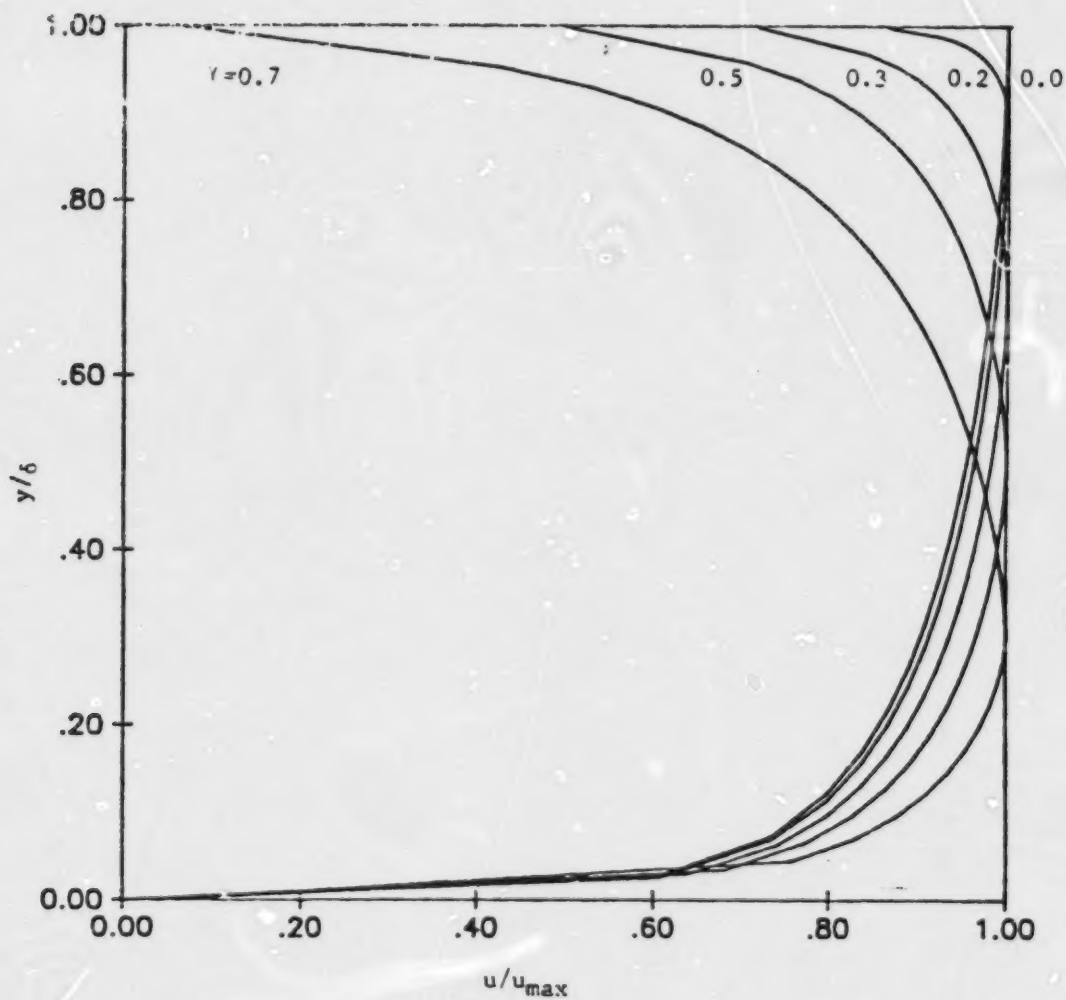


Fig. 5 Velocity profiles depending on interfacial shear stress

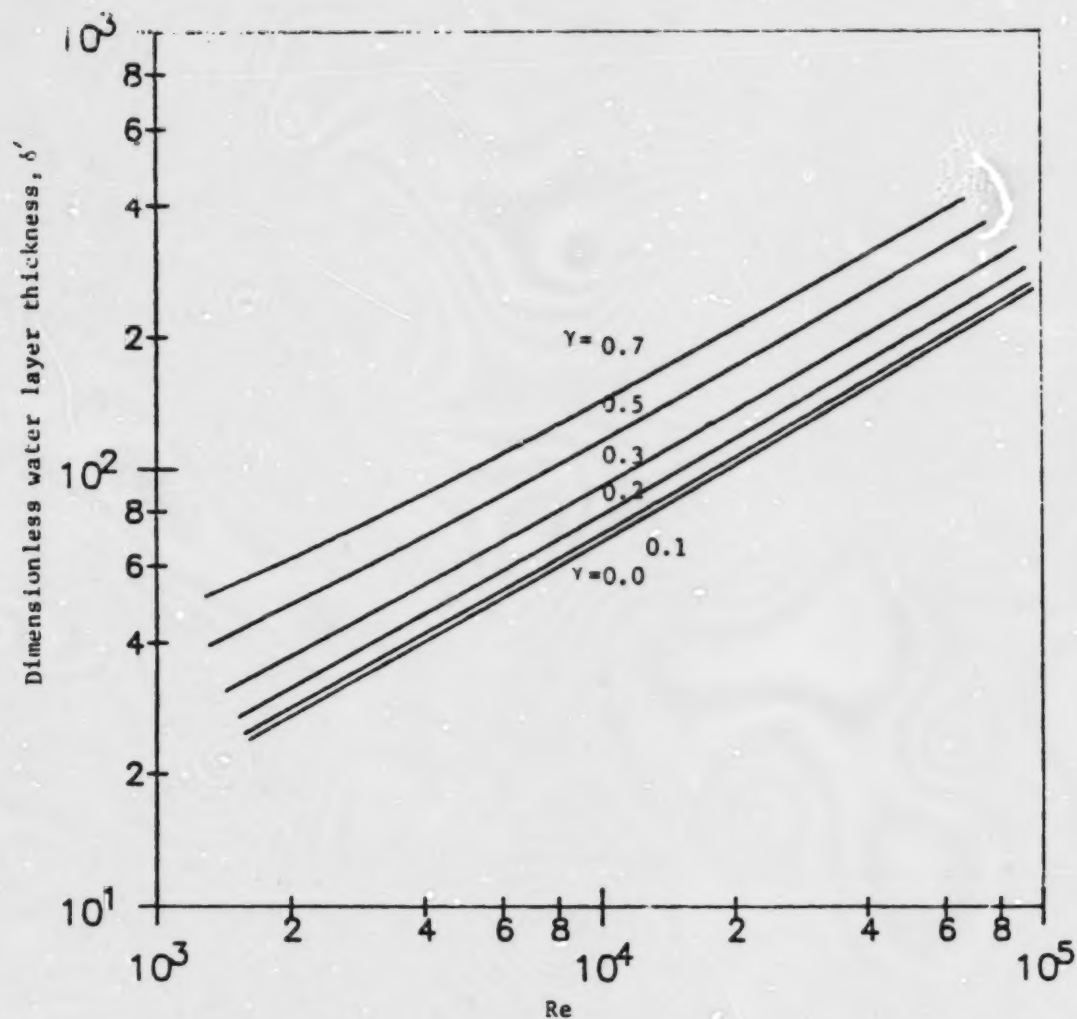


Fig. 6 Relation between  $\delta'$  and  $Re$  depending on interfacial shear stress

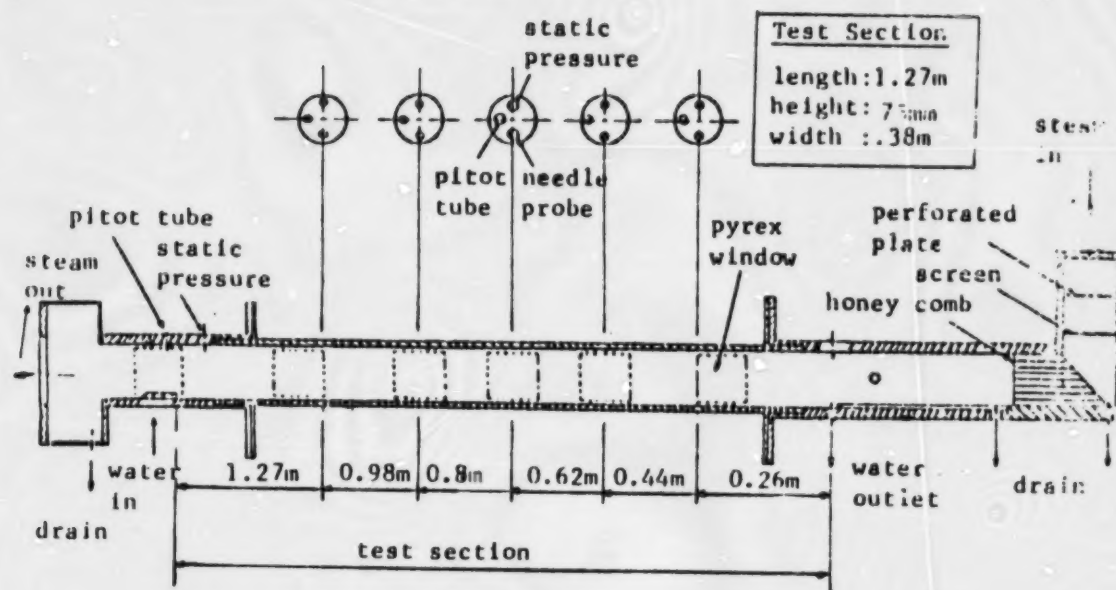


Fig. 7 Sketch of test section

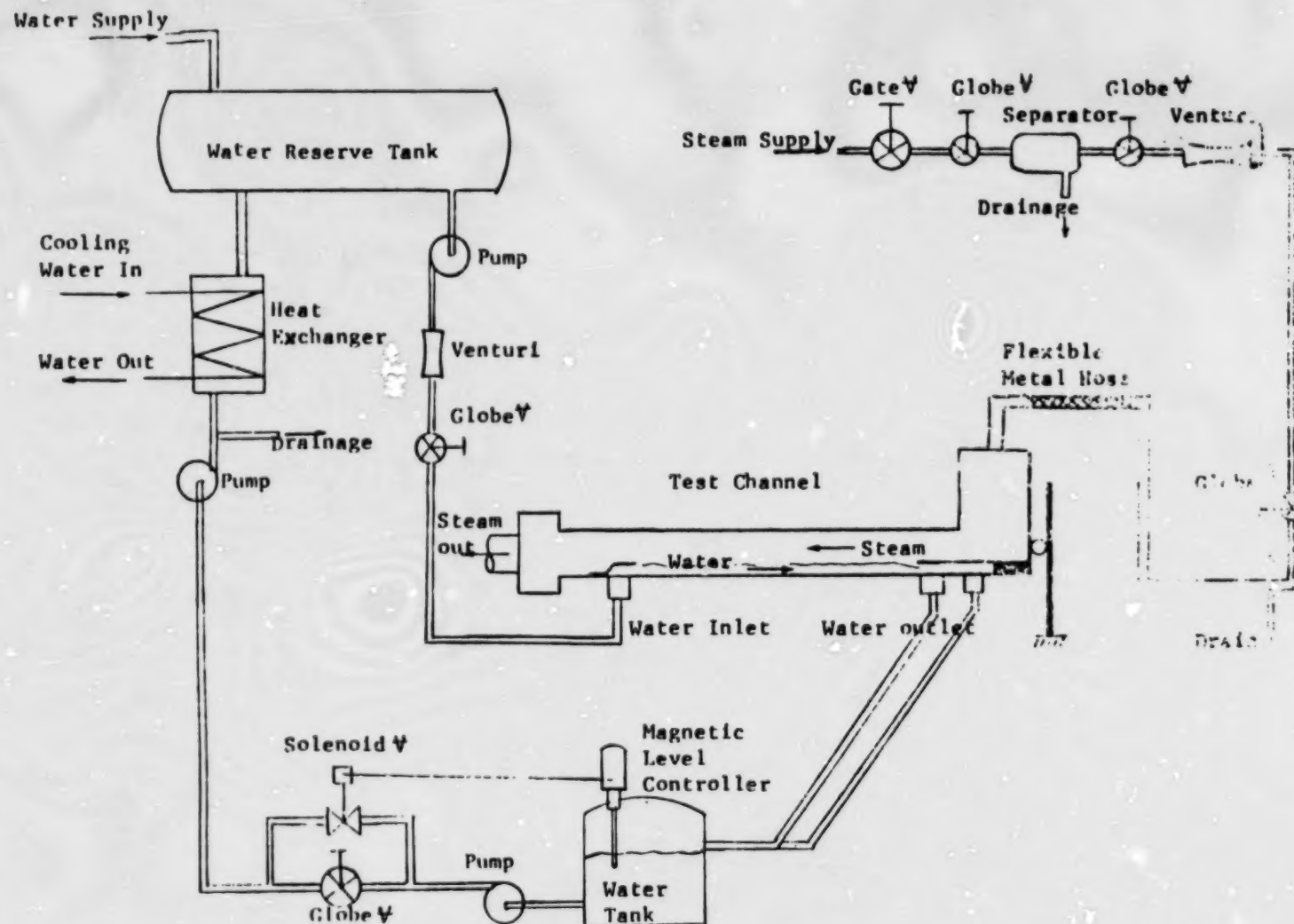


Fig.8 Overall system of test facility

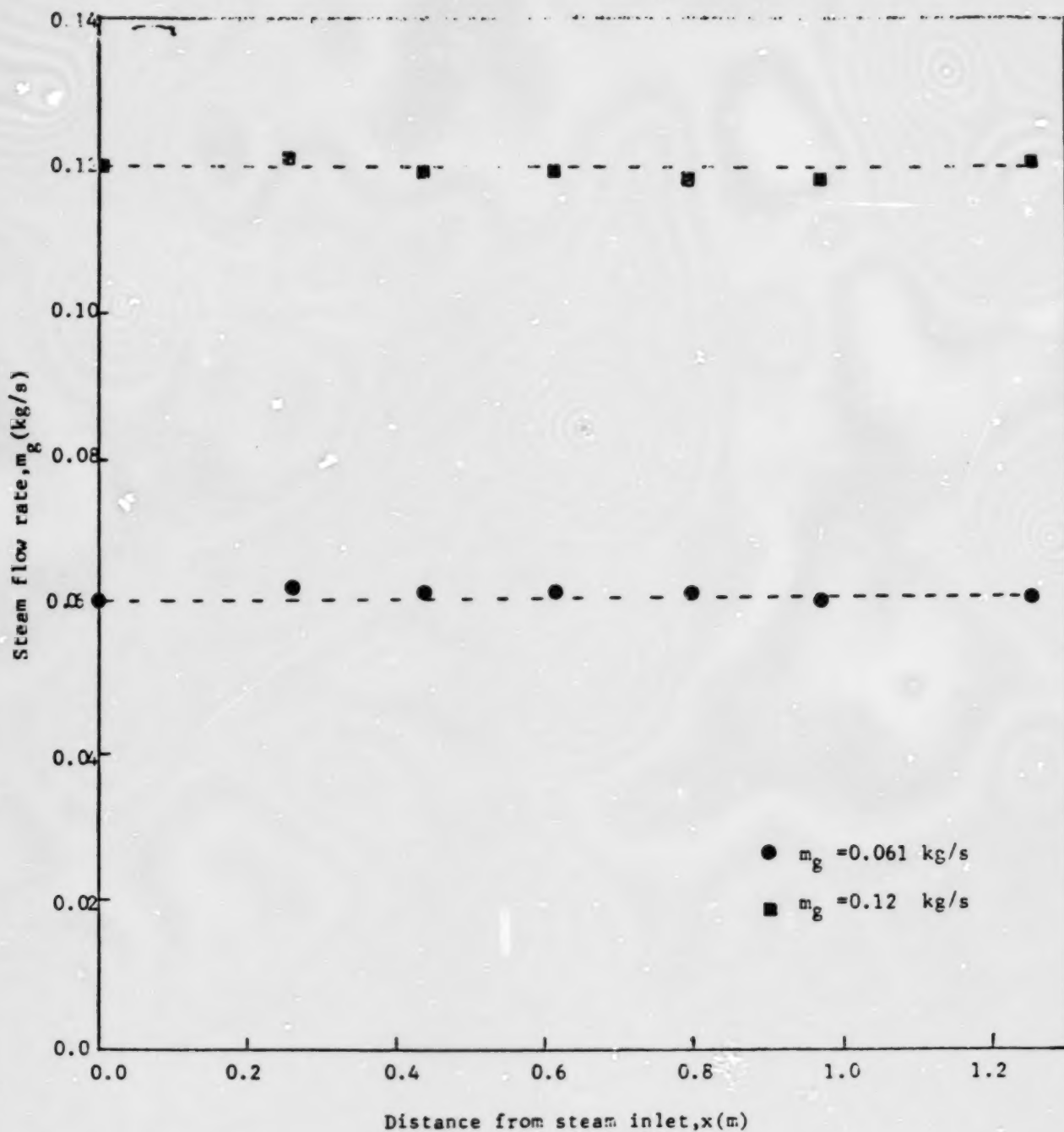


Figure. 9 Comparison between measured inlet steam flow rate and calculated local steam flow rate

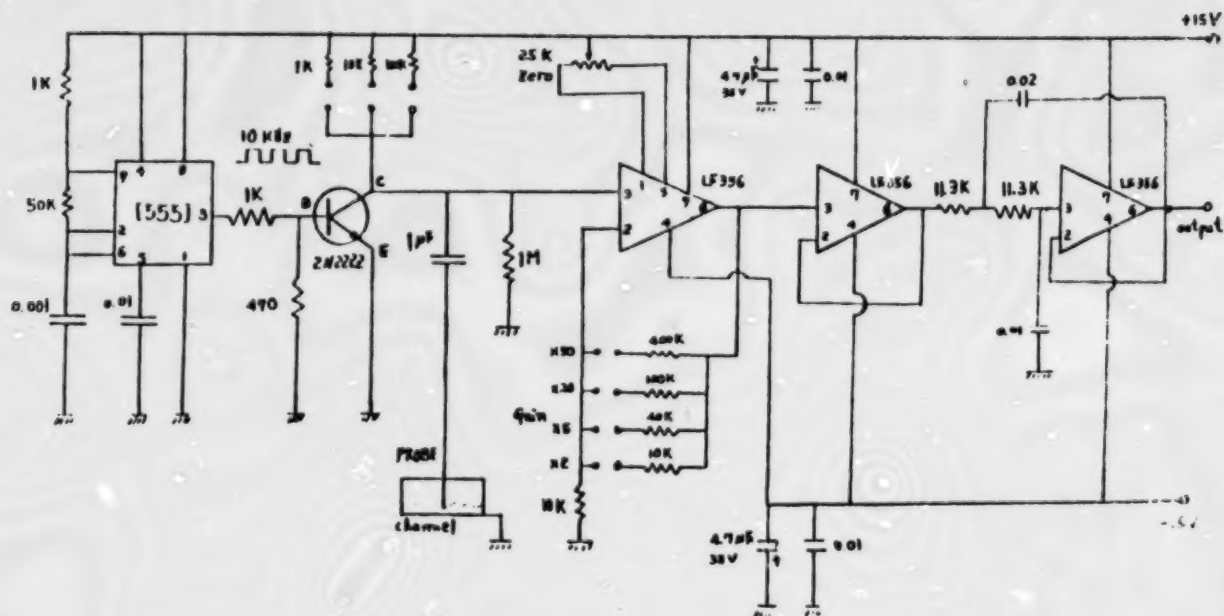
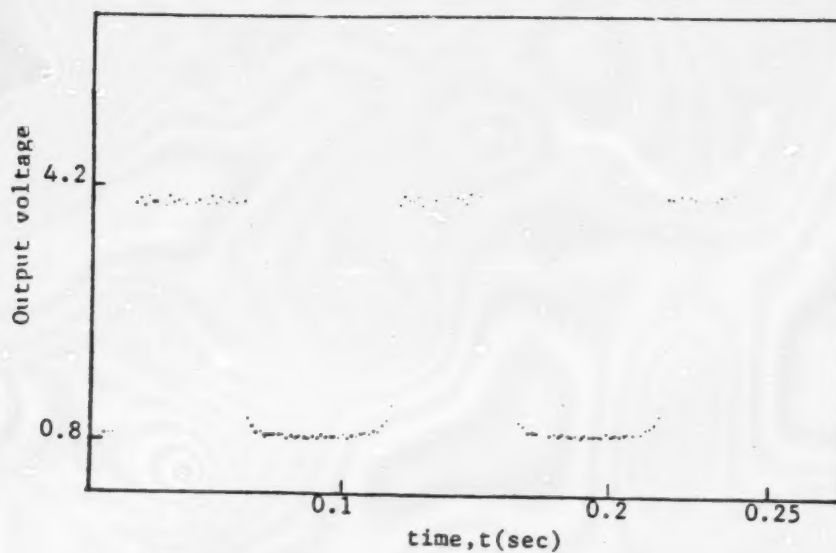
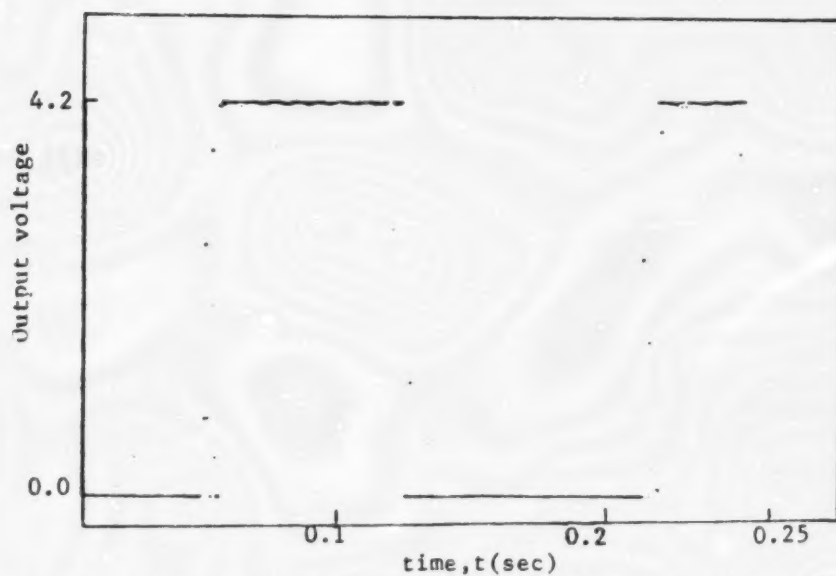


Fig. 10 A.C. probe circuitry



a) Signal from needle probe at gas-liquid interface



b) Signal from the contact of probe to solid surface

Fig. 11 Typical signal obtained from needle contact probe

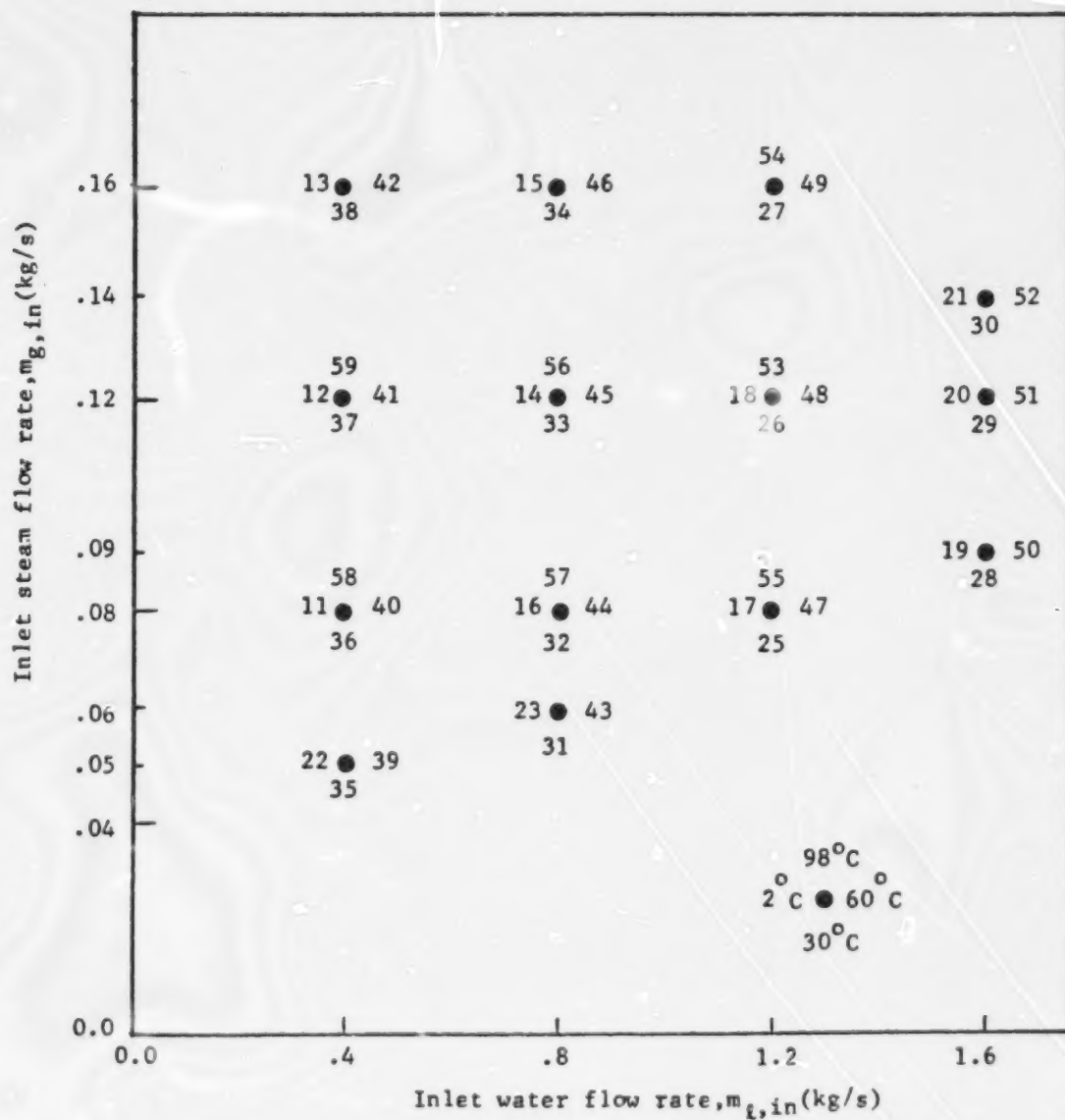


Fig. 12 Test matrix for  $\theta=4$ ,  $B/H=5$  (number is test number)

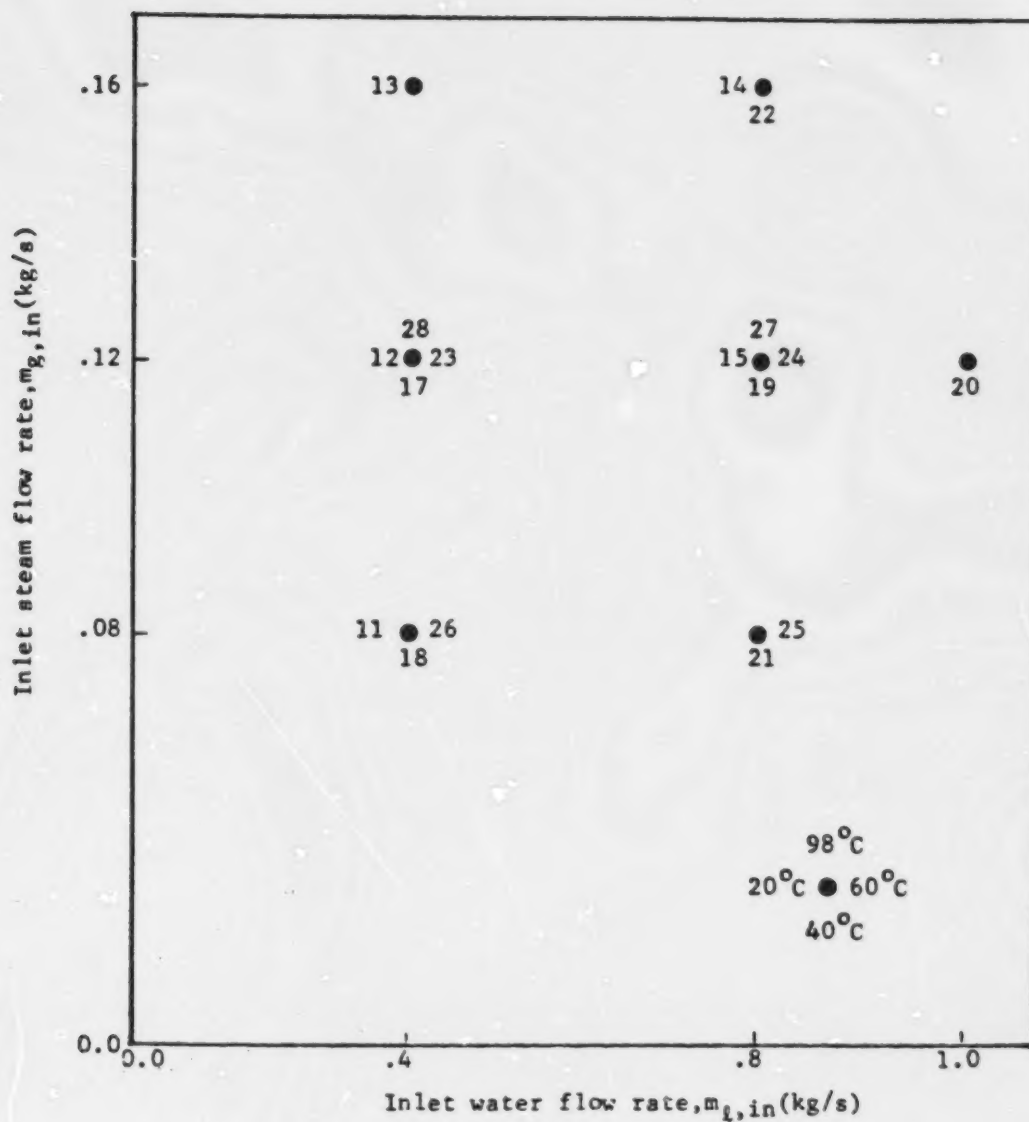


Fig. 13 Test matrix for  $\theta=30$ ,  $B/H=5$  (number is test number)

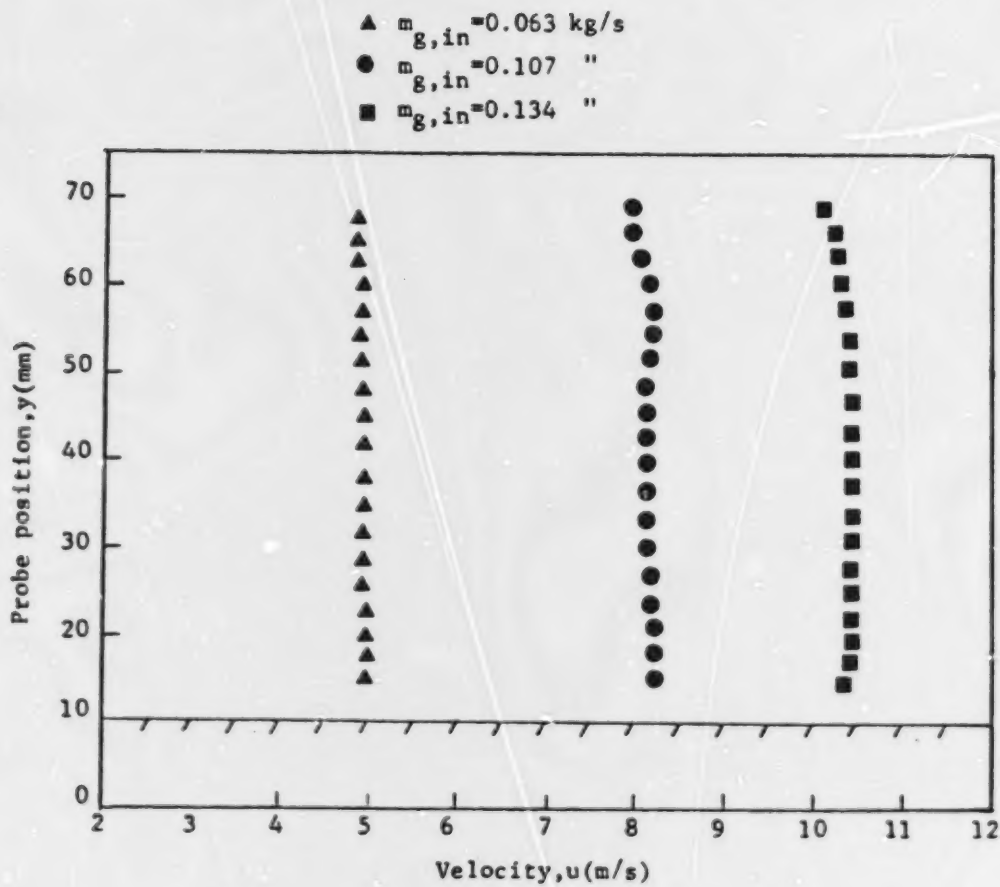


Fig. 14 Steam velocity profiles at steam entrance

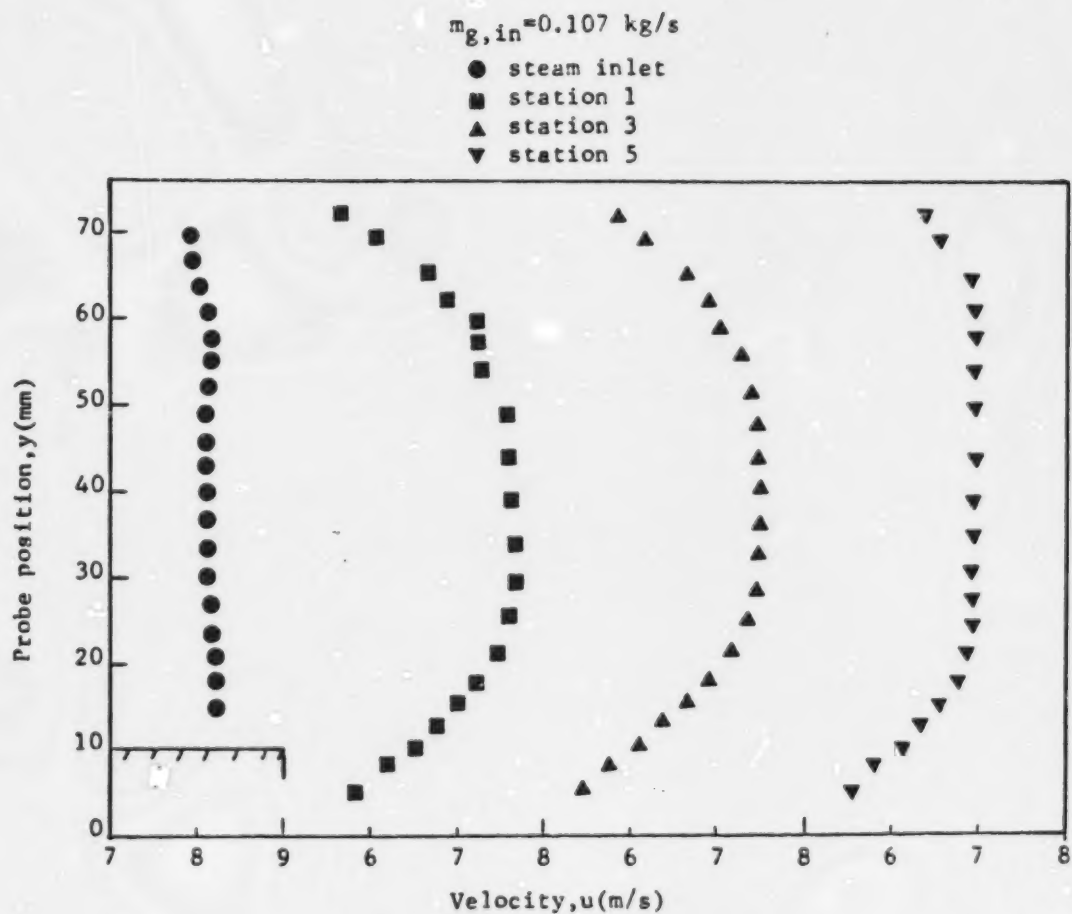


Fig. 15 Development of velocity profiles in test section with no water flow

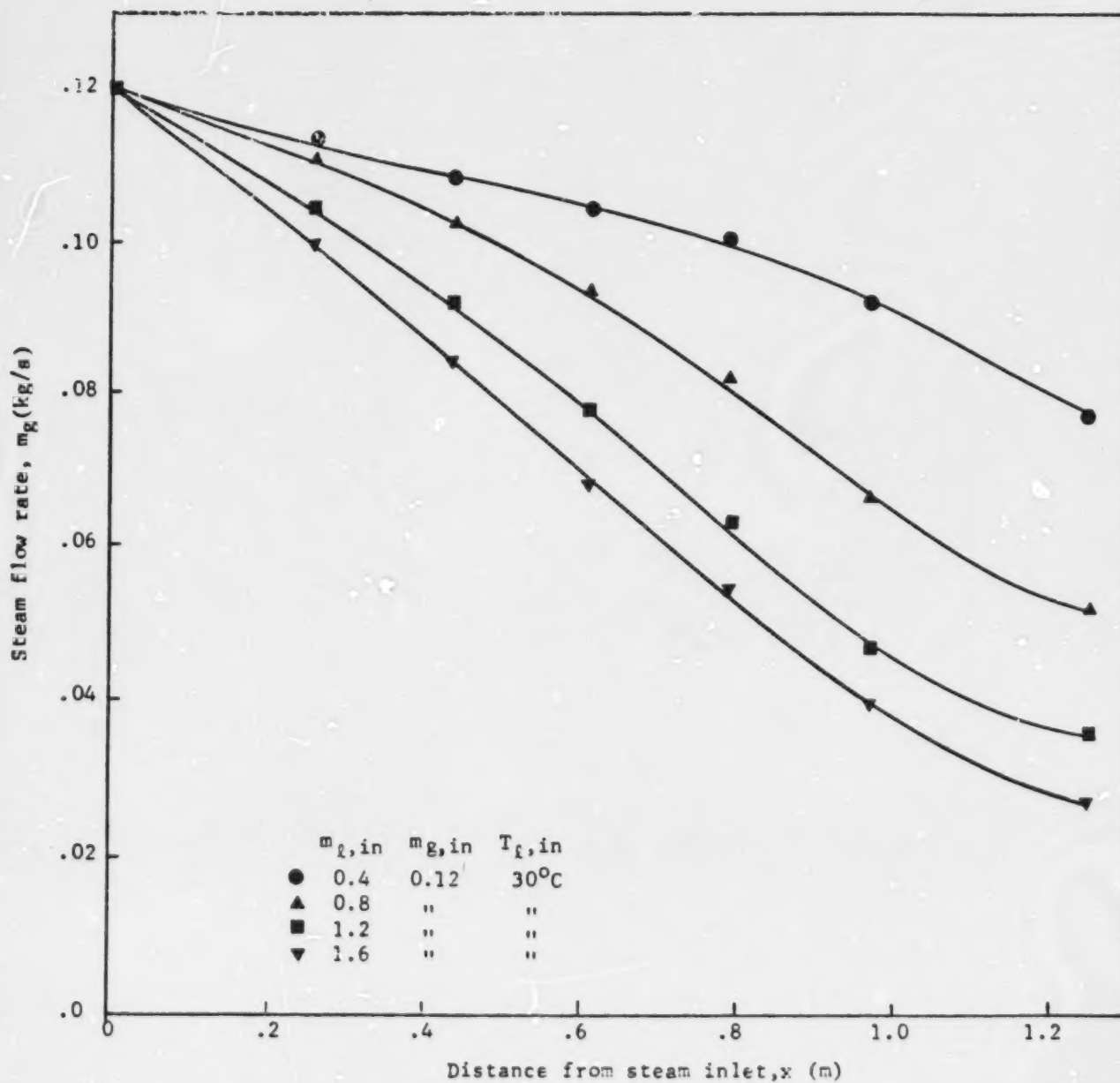


Fig. 16 Axial steam flow rate profiles as a function of inlet water flow rate

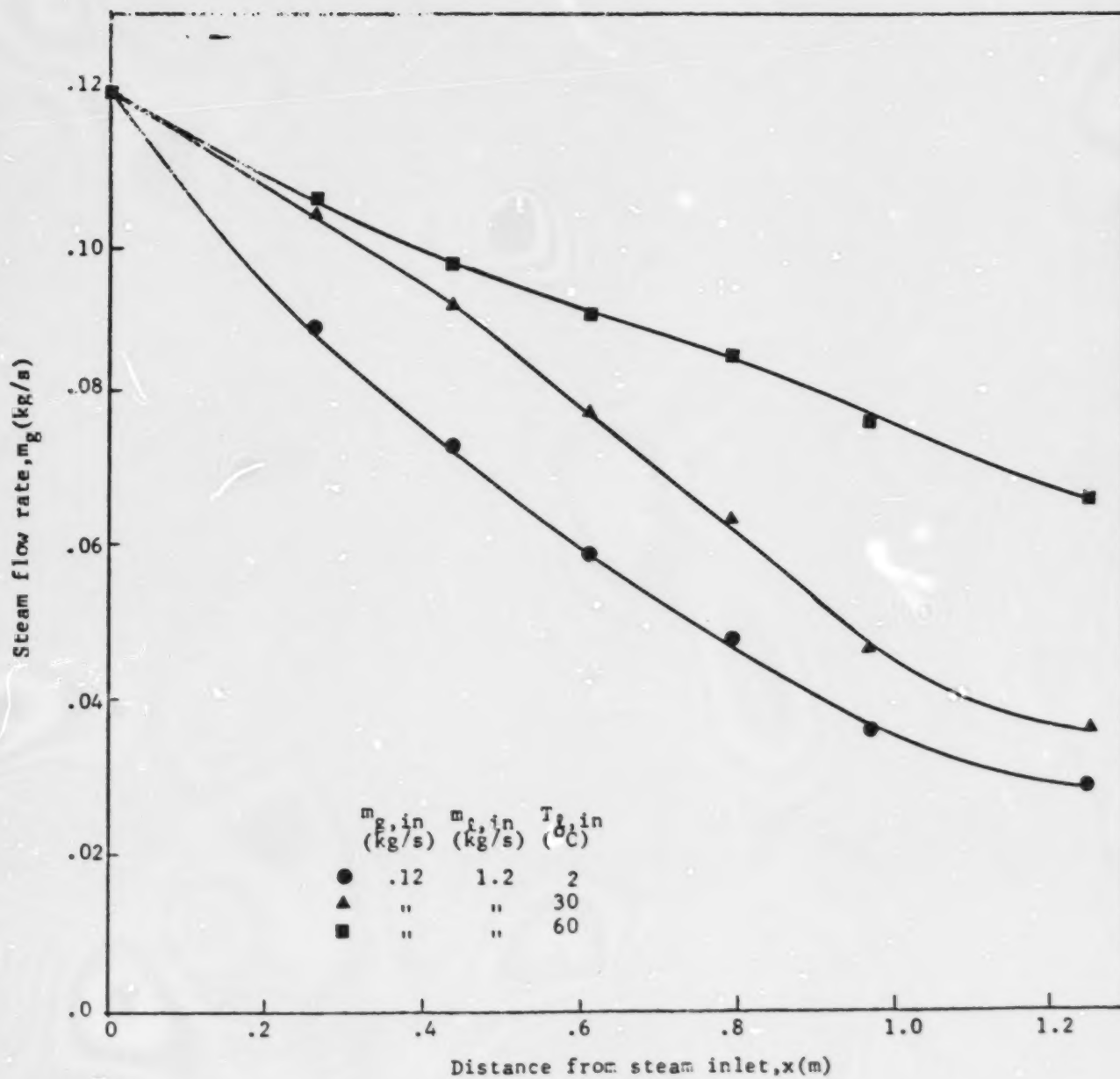


Fig. 17 Axial steam flow rate profiles as a function of inlet water temperature

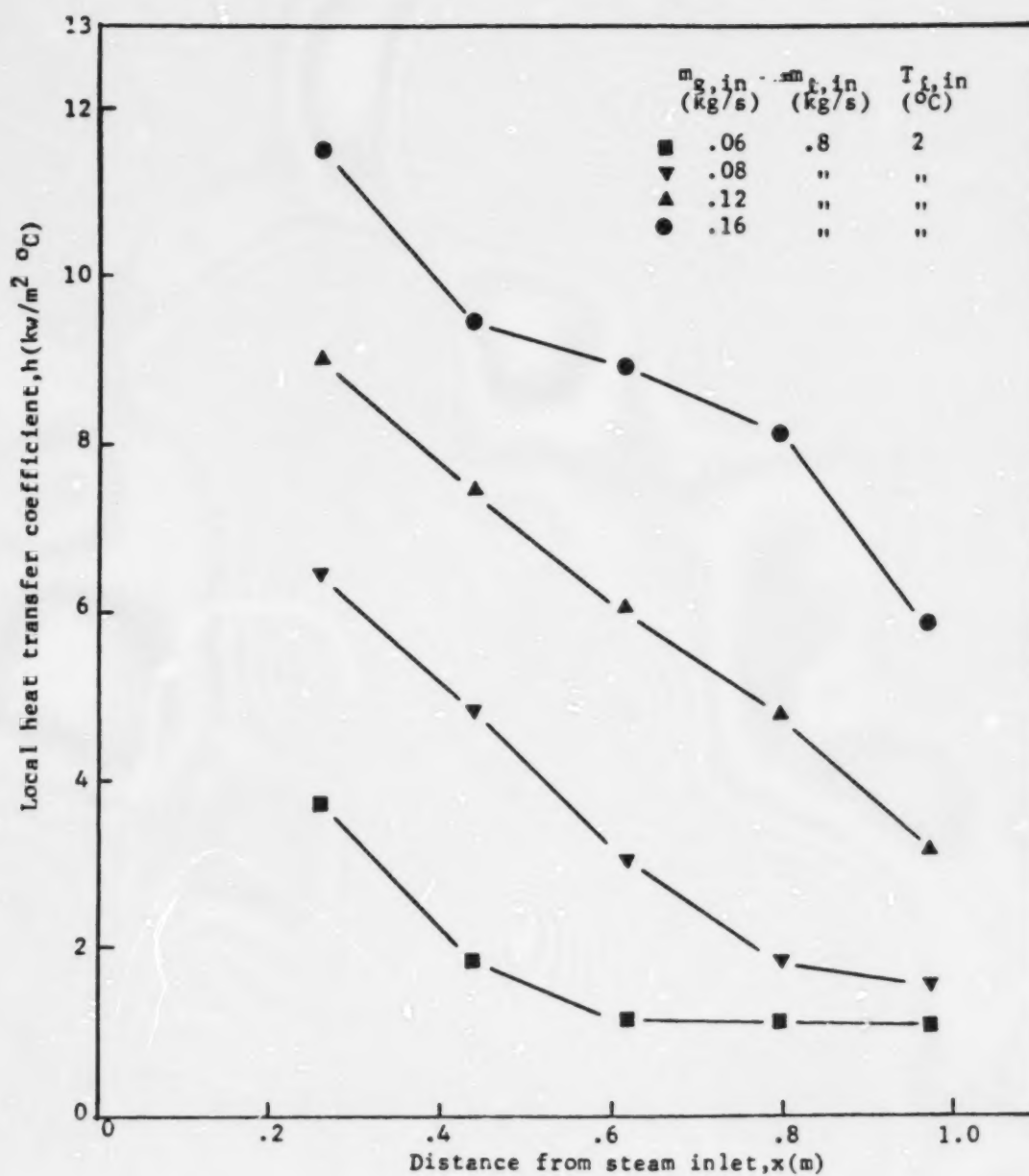


Fig. 18 Axial heat transfer coefficient profiles as a function of inlet steam flow rate

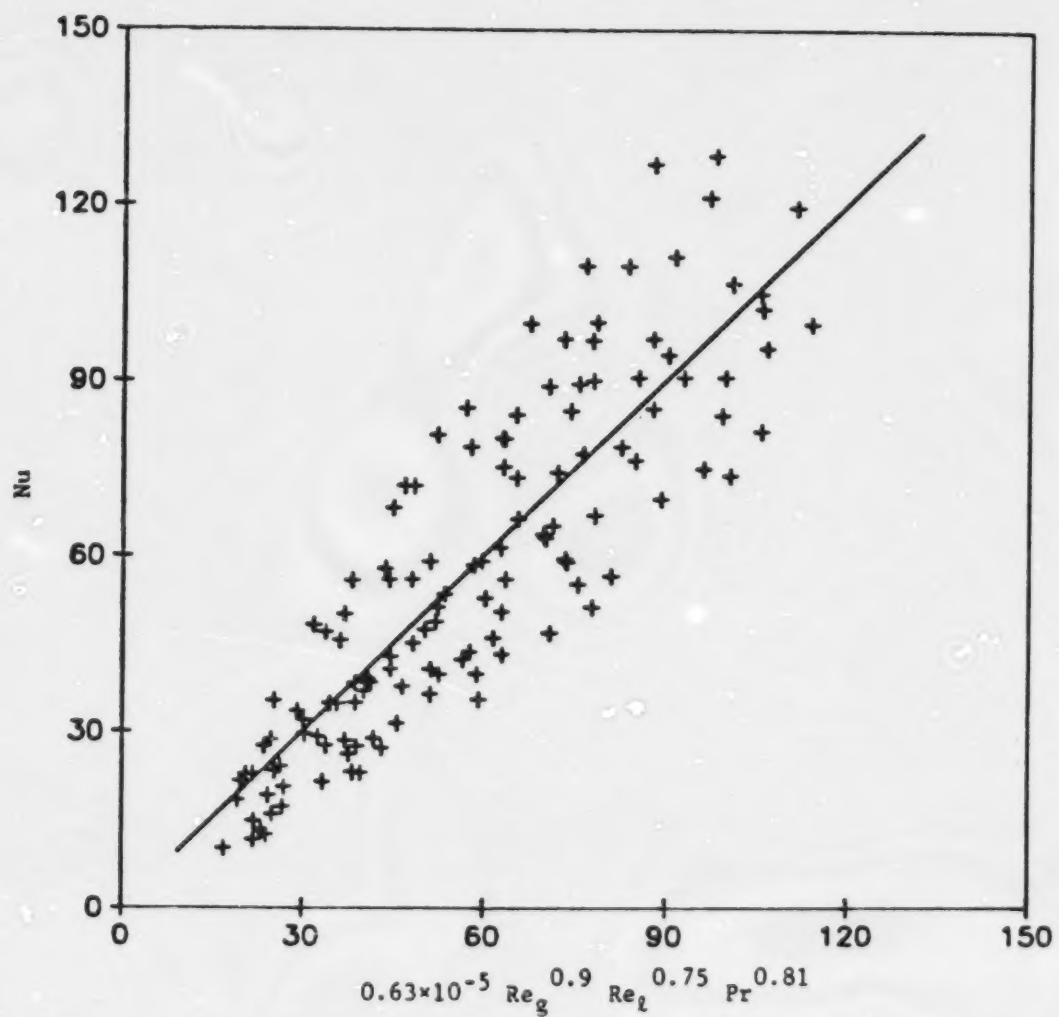


Fig. 19 Experimental correlation for  $\theta=4$ ,  $B/H=5$

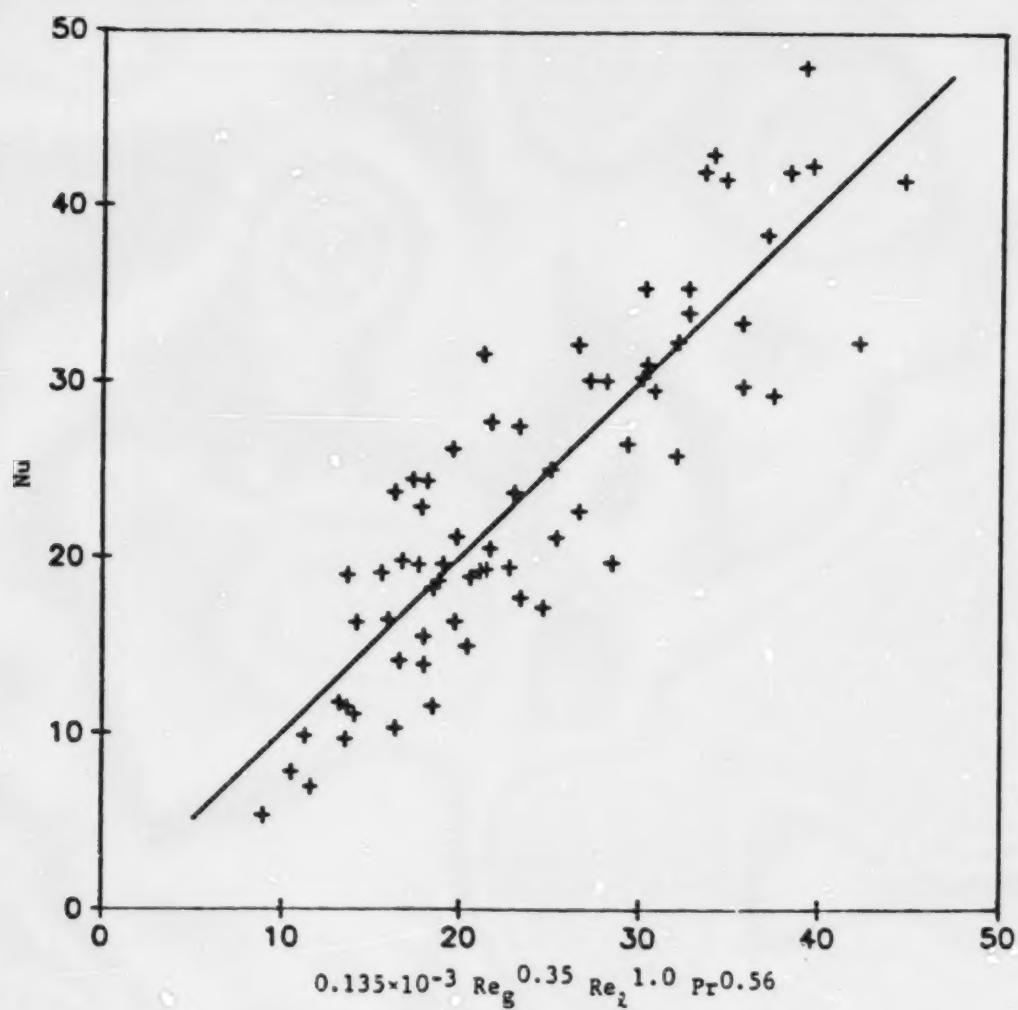


Fig. 20 Experimental correlation for  $\theta = 30$ ,  $B/H = 5$

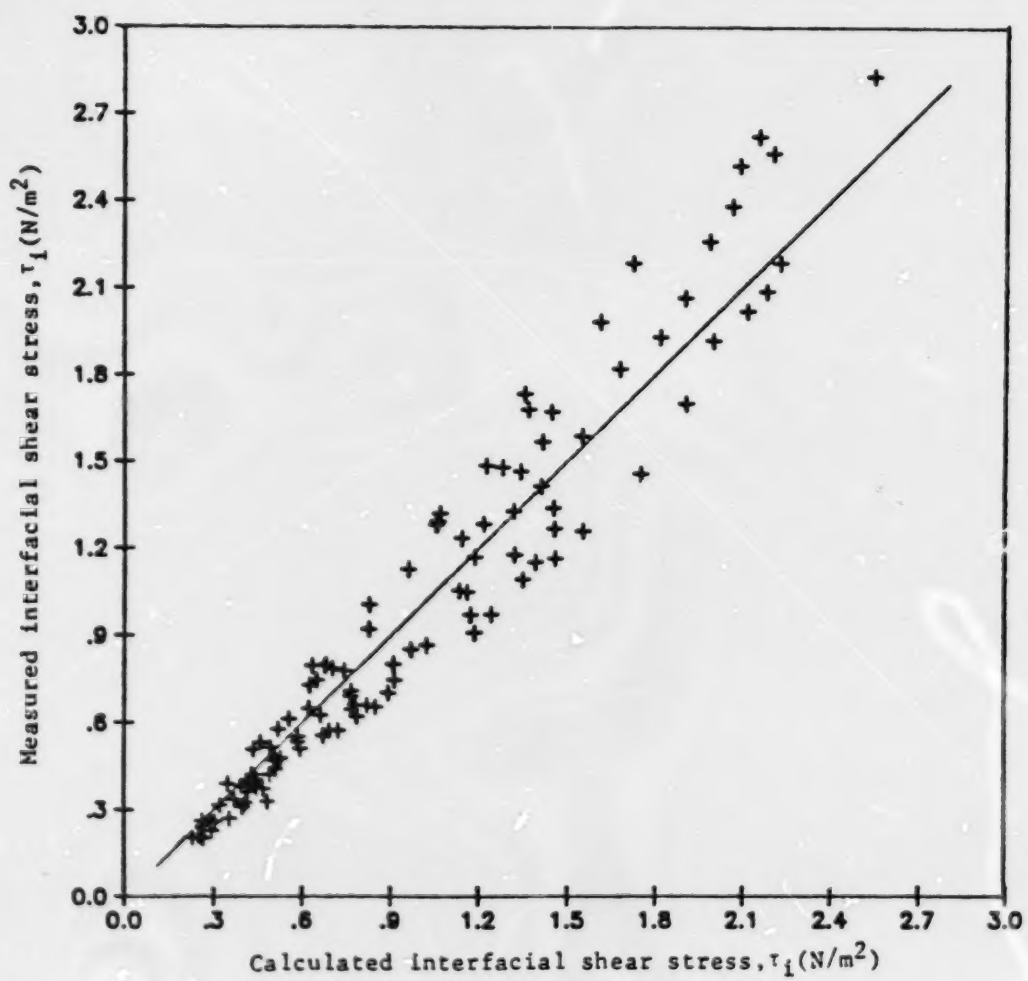


Fig. 21 Comparison of measured interfacial shear stress with the calculated value for  $\theta=4$ ,  $B/H=5$

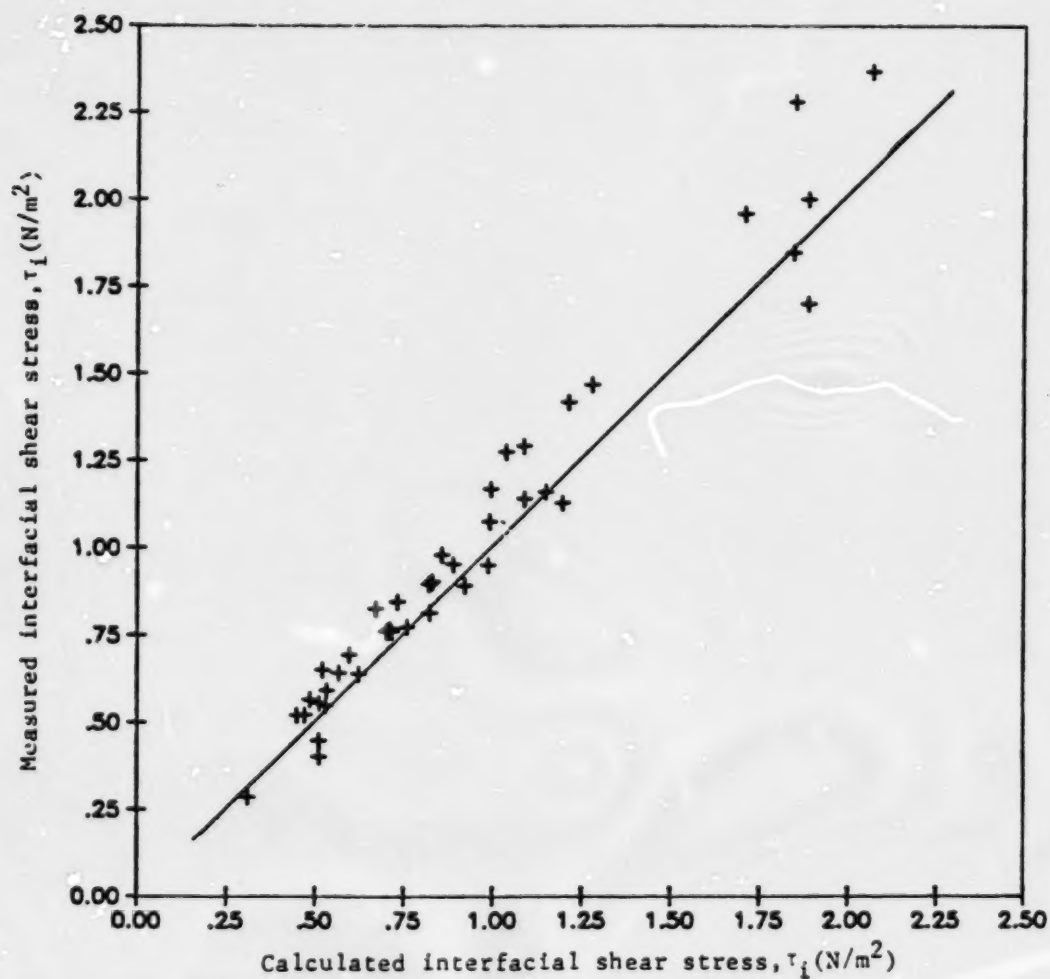


Fig. 22 Comparison of measured interfacial shear stress with the calculated value for  $\theta=30$ ,  $B/H=5$

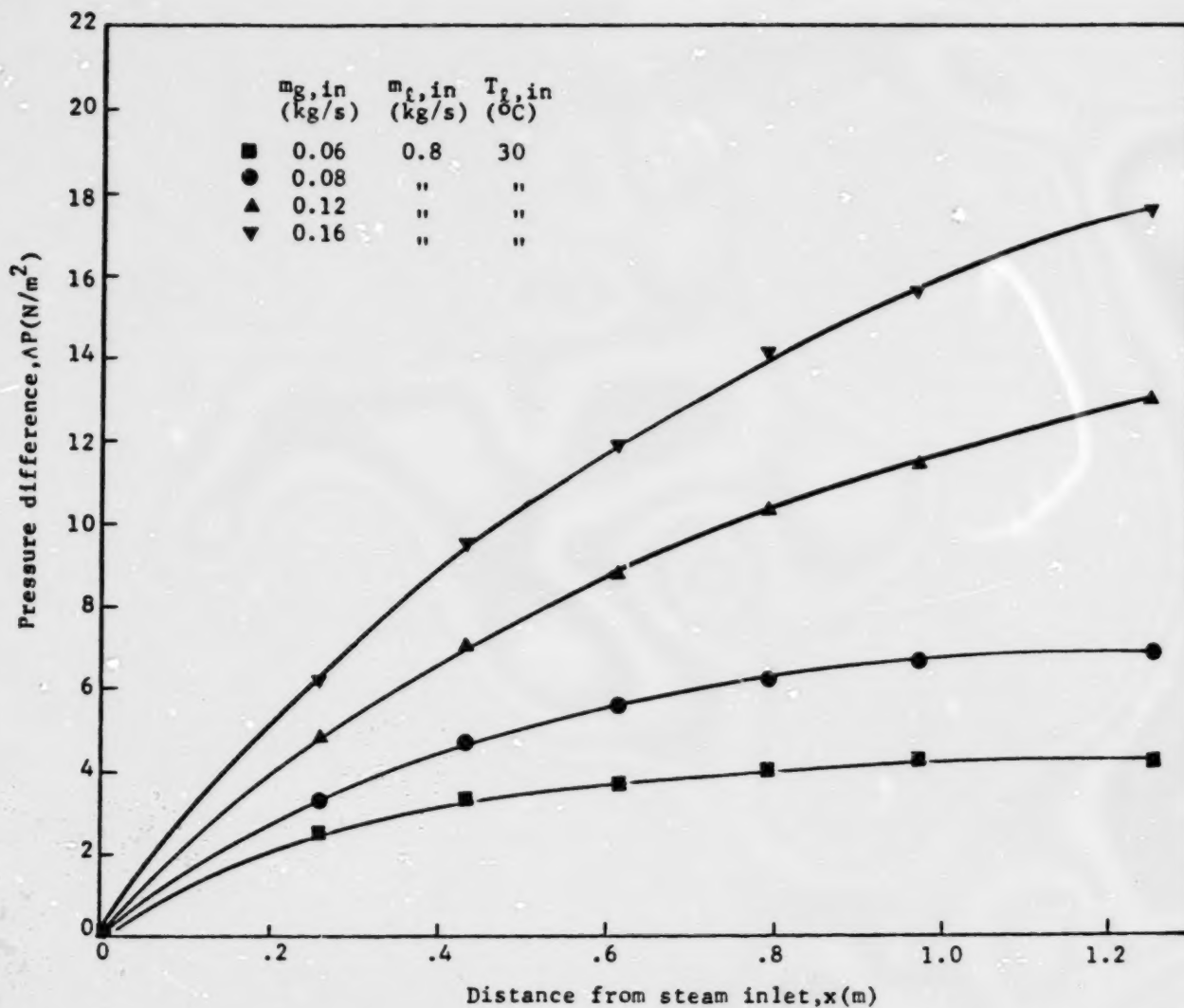


Fig. 23 Axial distribution of pressure difference as a function of inlet steam flow rate

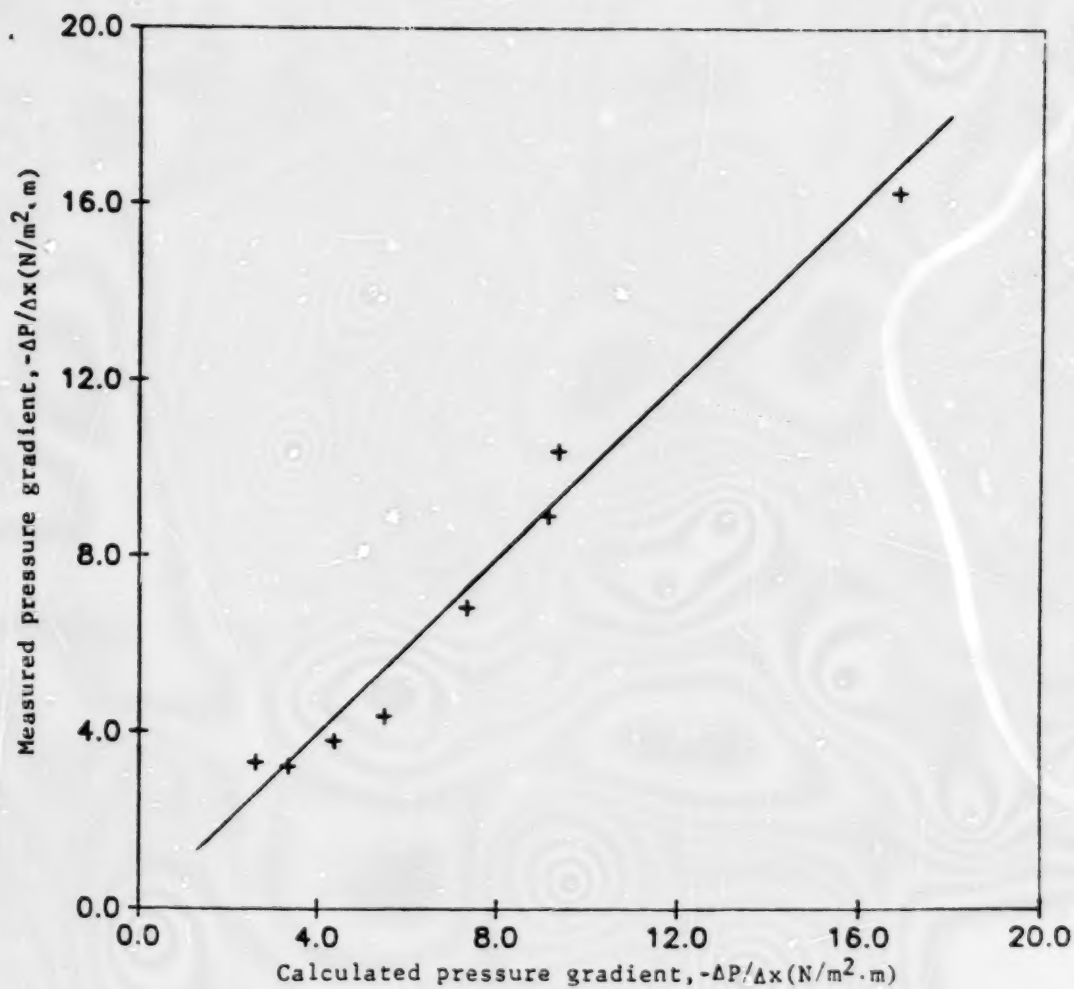


Fig. 24 Comparison of measured pressure gradient with the calculated value in the absence of condensation

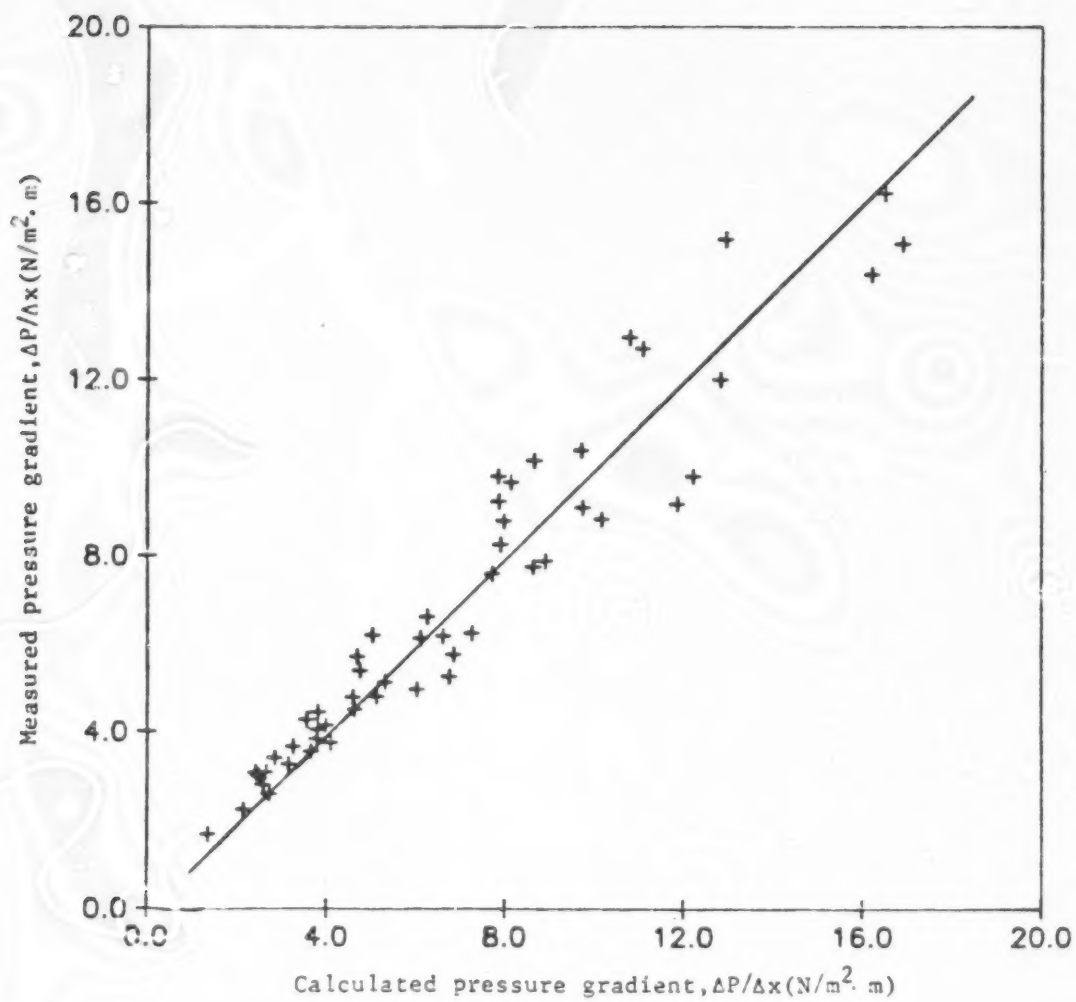


Fig. 25 Comparison of measured pressure gradient with the calculated value for  $\theta=4$ ,  $B/H=5$

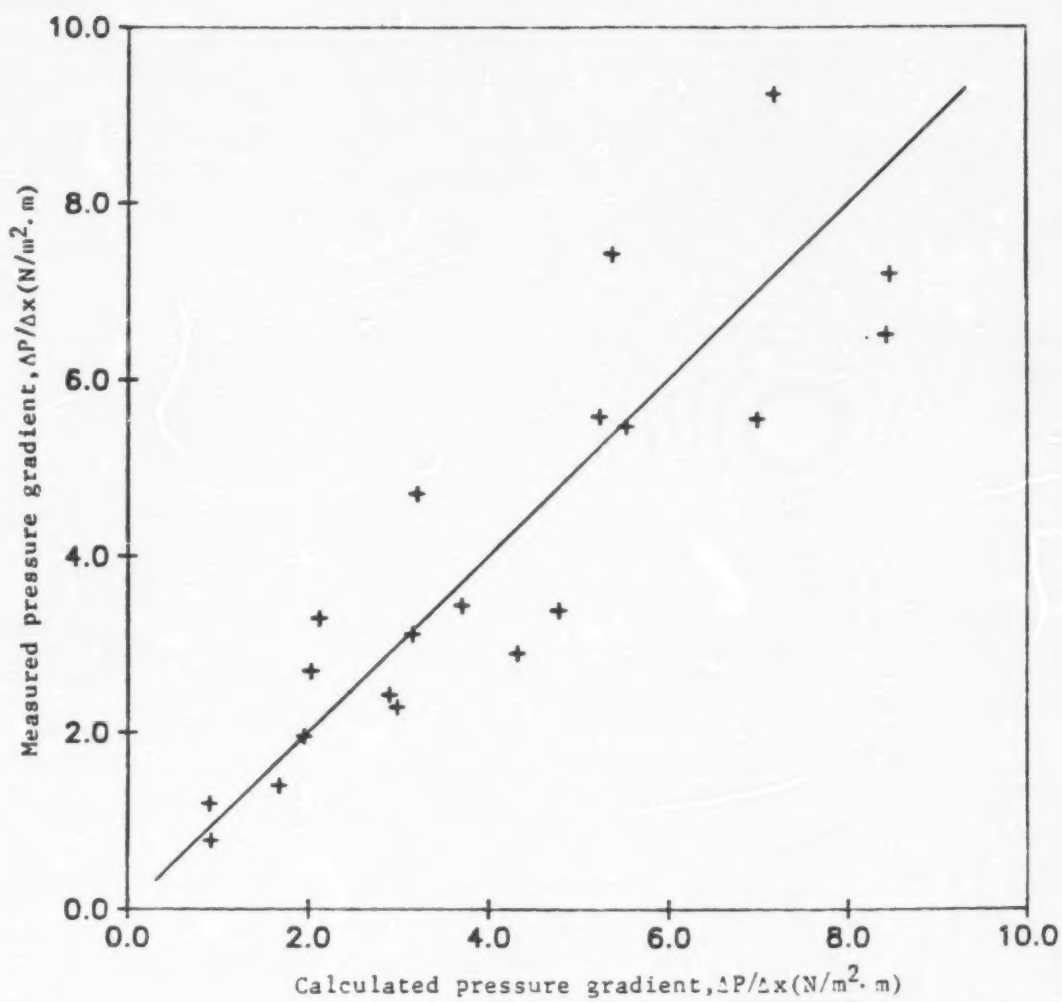


Fig. 26 Comparison of measured pressure gradient with the calculated value for  $\theta=30$ ,  $B/H=5$

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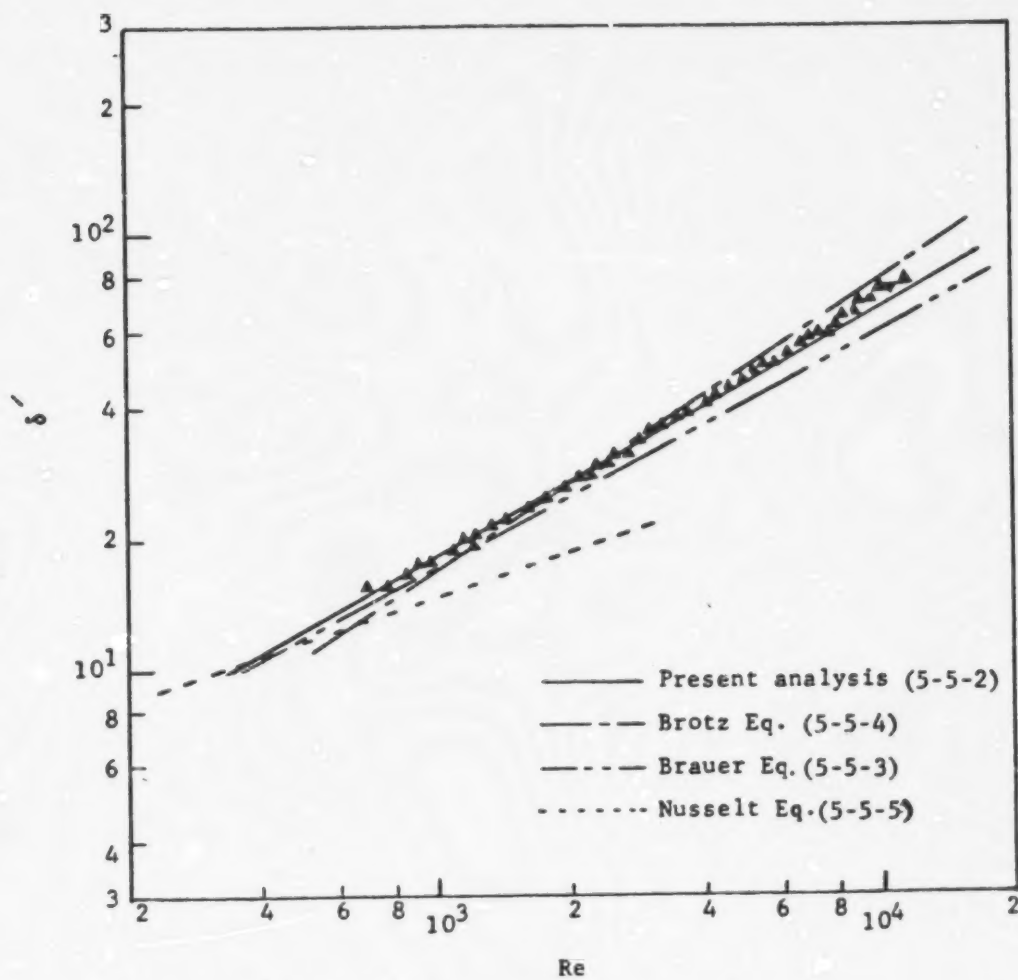


Fig. 27 Comparison of measured water layer thickness with theoretical models without interfacial shear stress

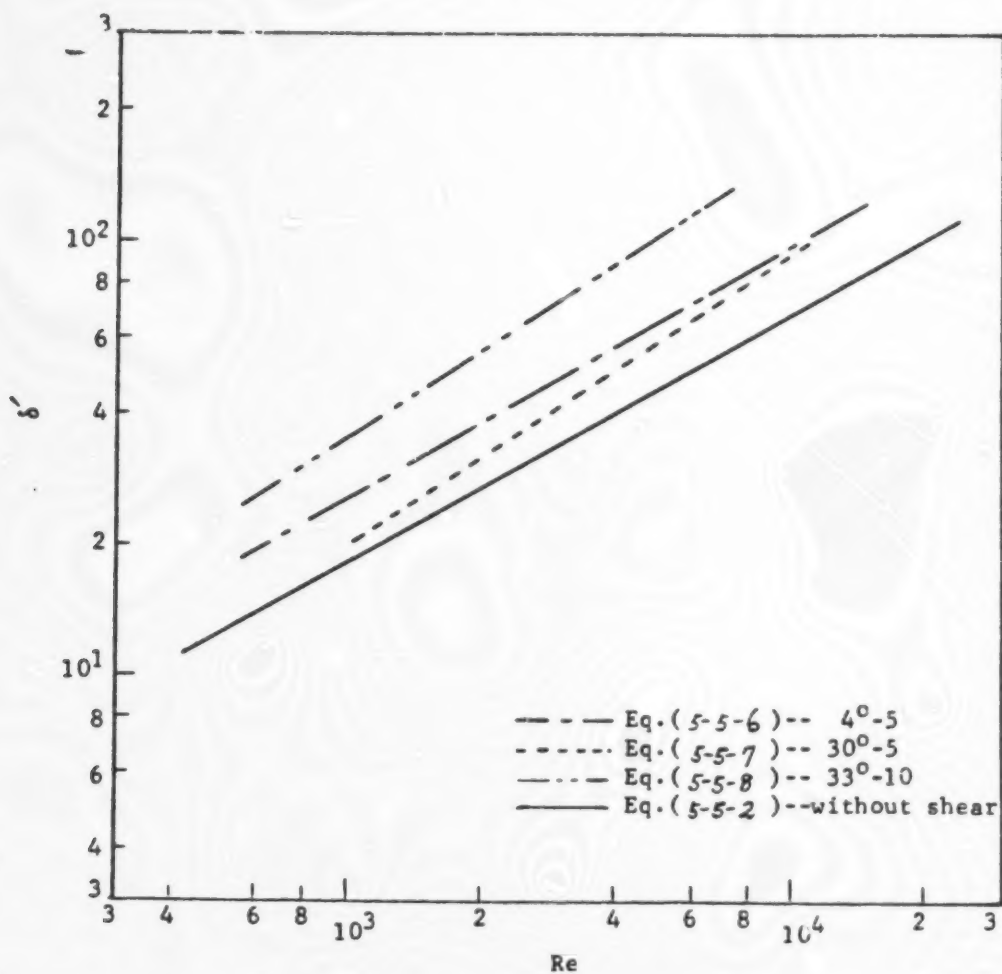


Fig. 28 Experimental correlation for water layer thickness for different geometries with interfacial shear stress

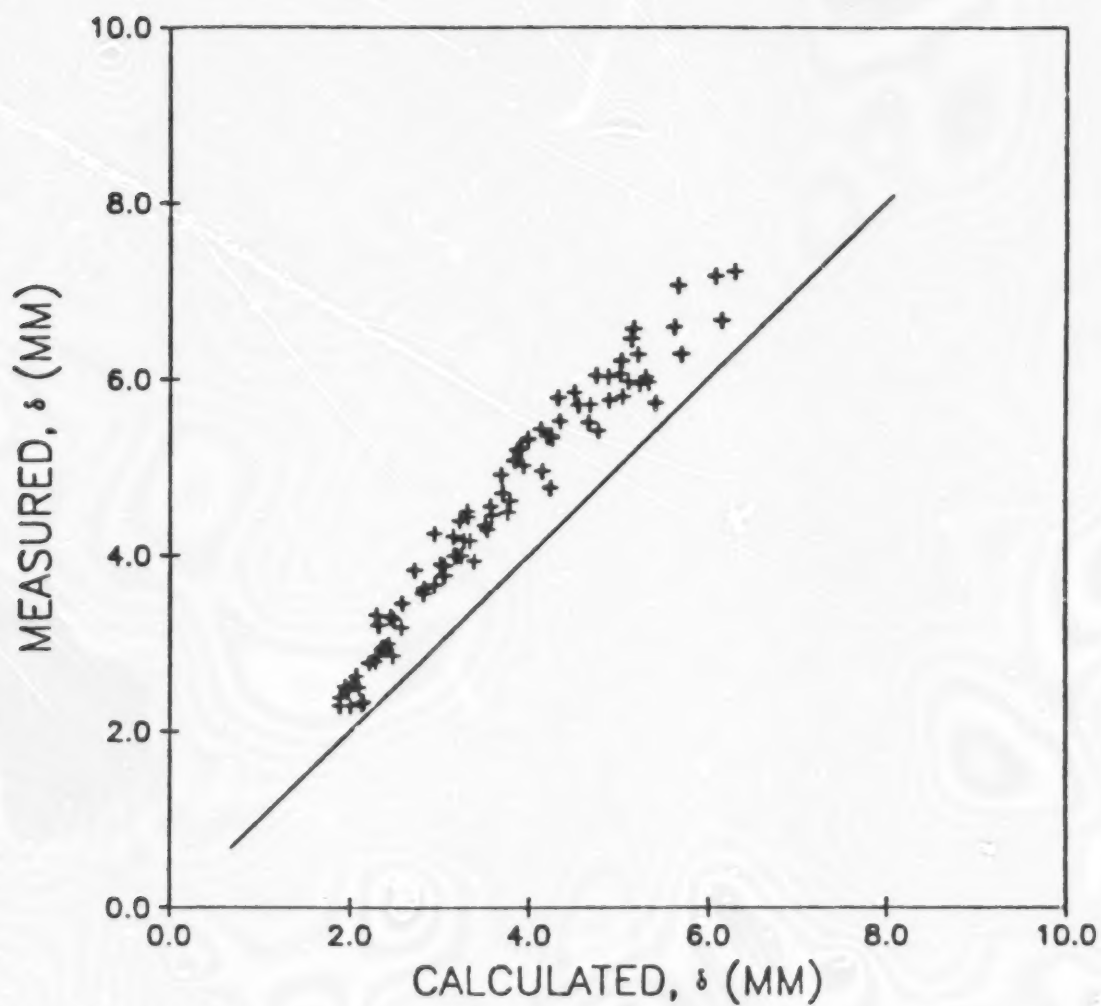


Fig. 29 Comparison of measured mean water layer thickness with the calculated value with interfacial shear stress

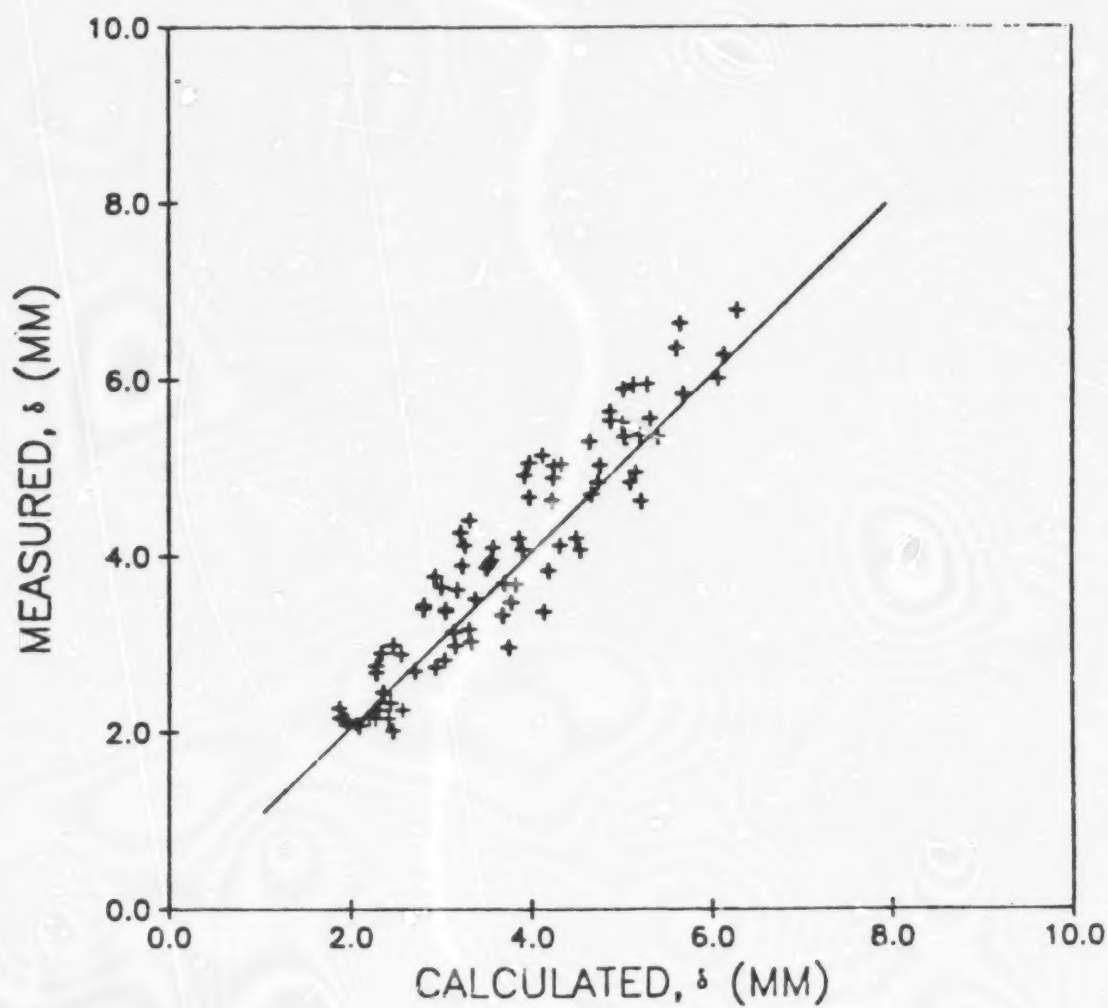


Fig. 30 Comparison of measured thickness of wave trough with the calculated value with interfacial shear stress

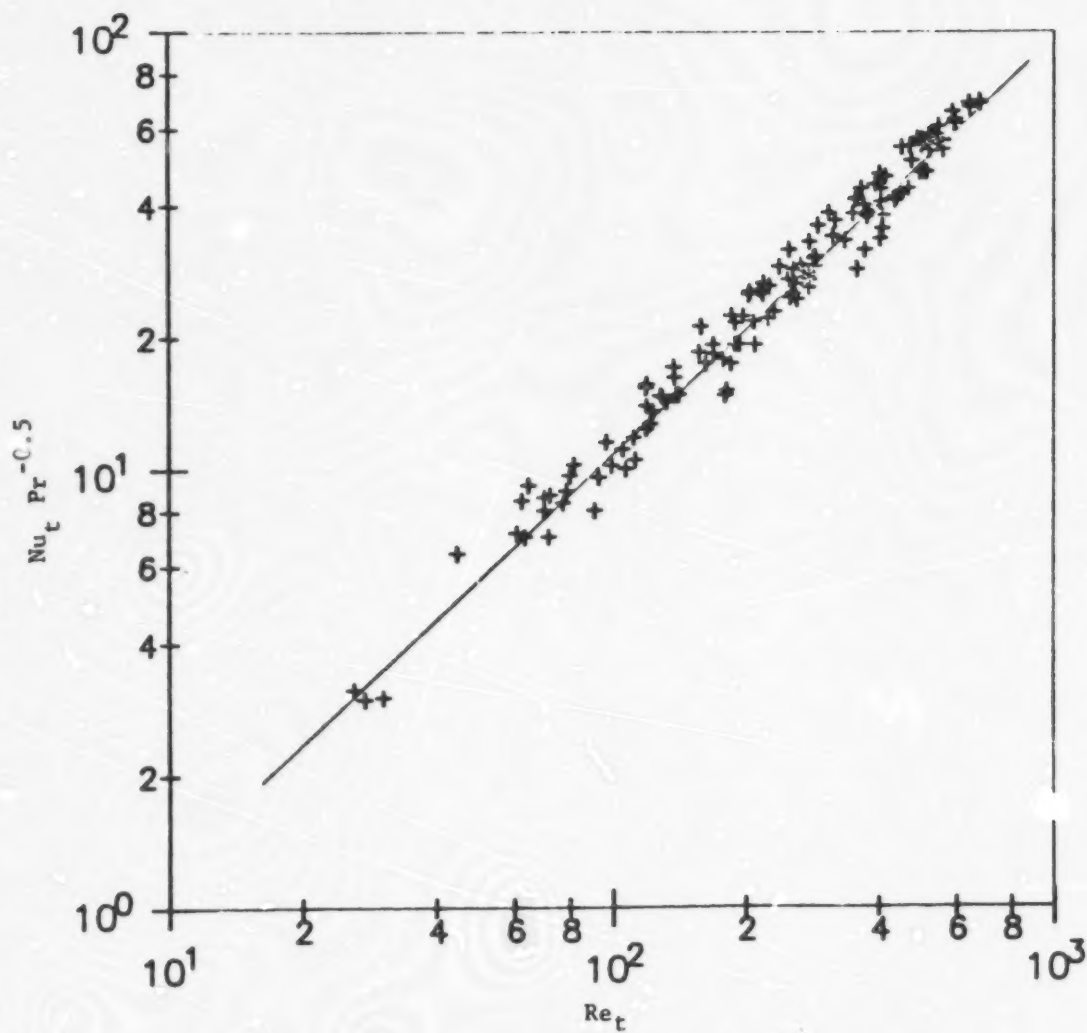


Fig. 31 Experimental correlation for  $\theta=4$ ,  $B/H=5$

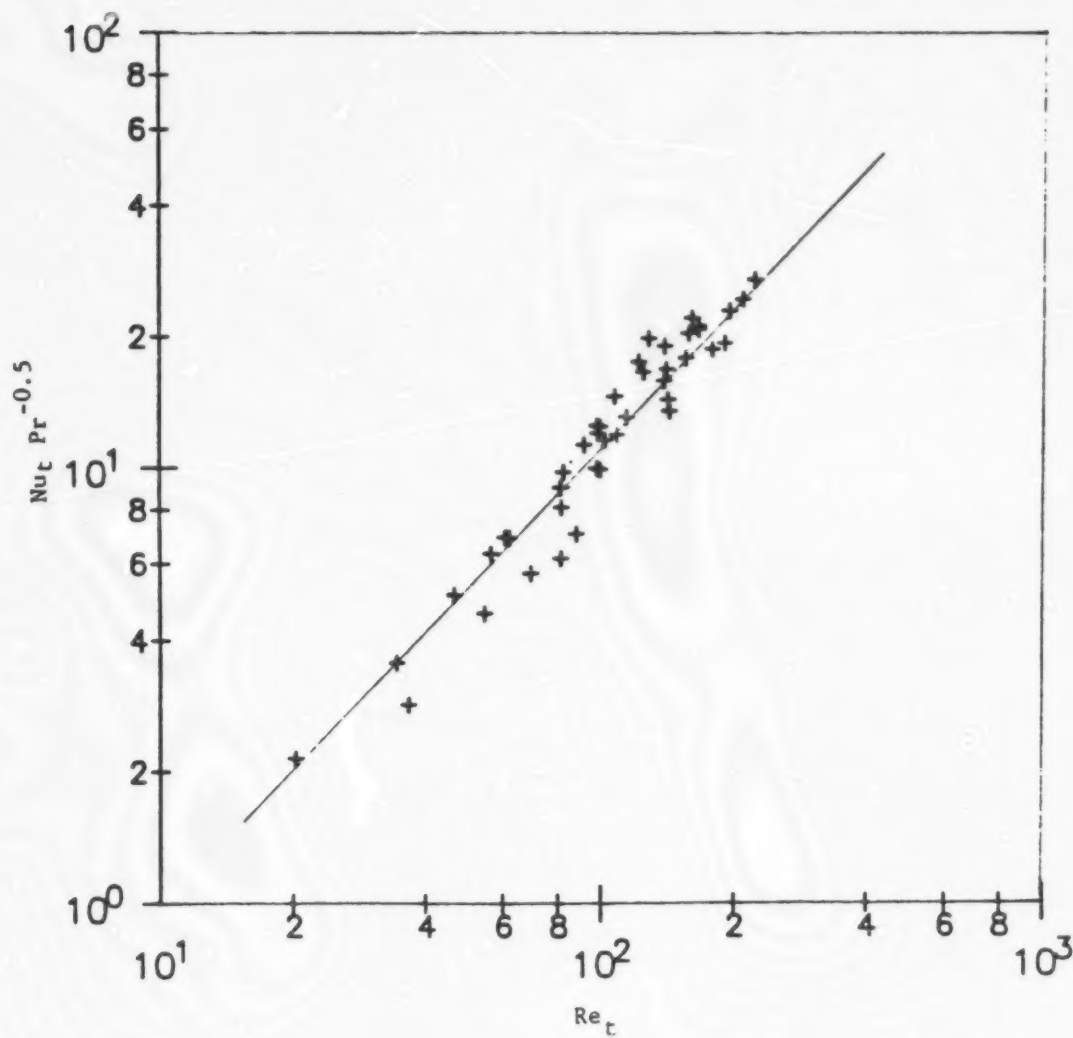


Fig. 32 Experimental correlation for  $\theta=30$ ,  $B/H=5$

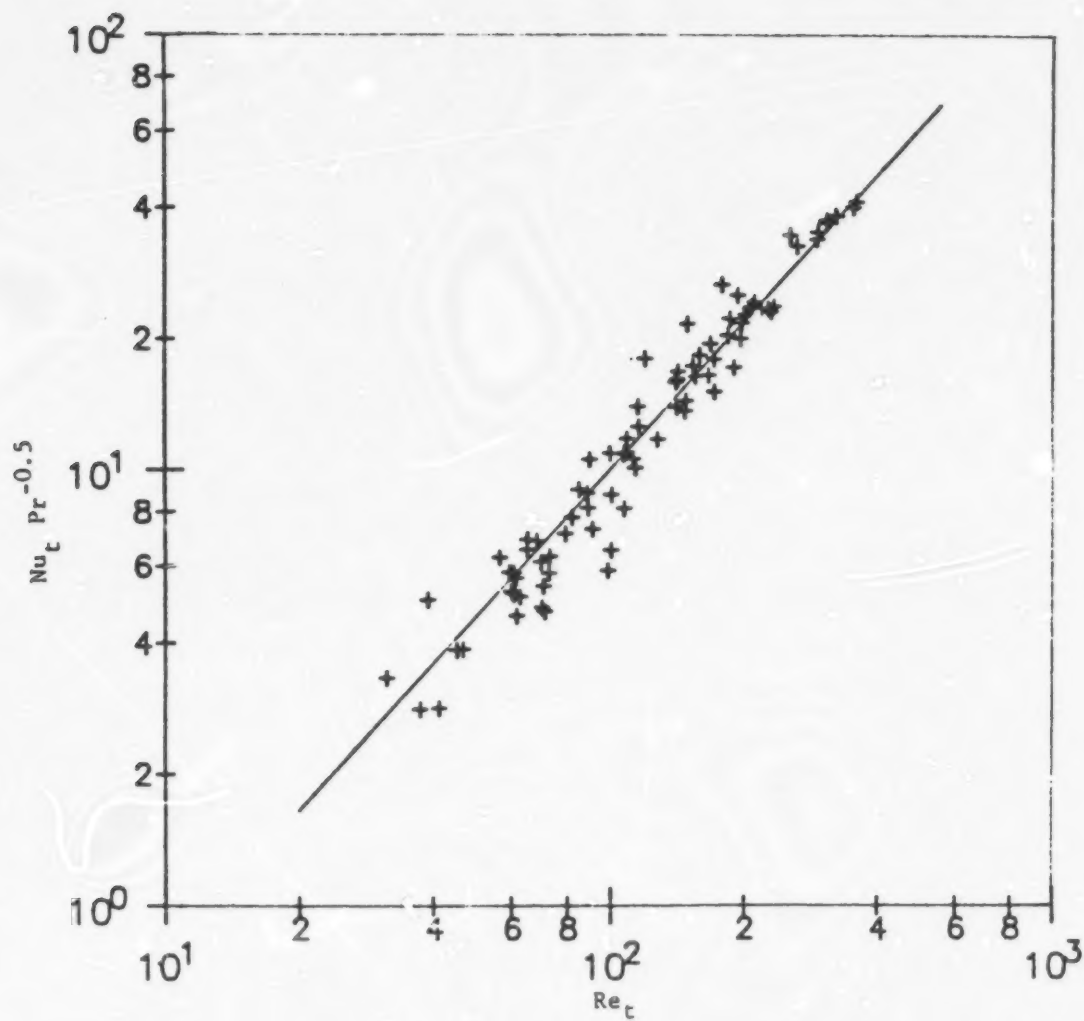


Fig. 33 Experimental correlation for  $\theta=33$ ,  $B/H=10$

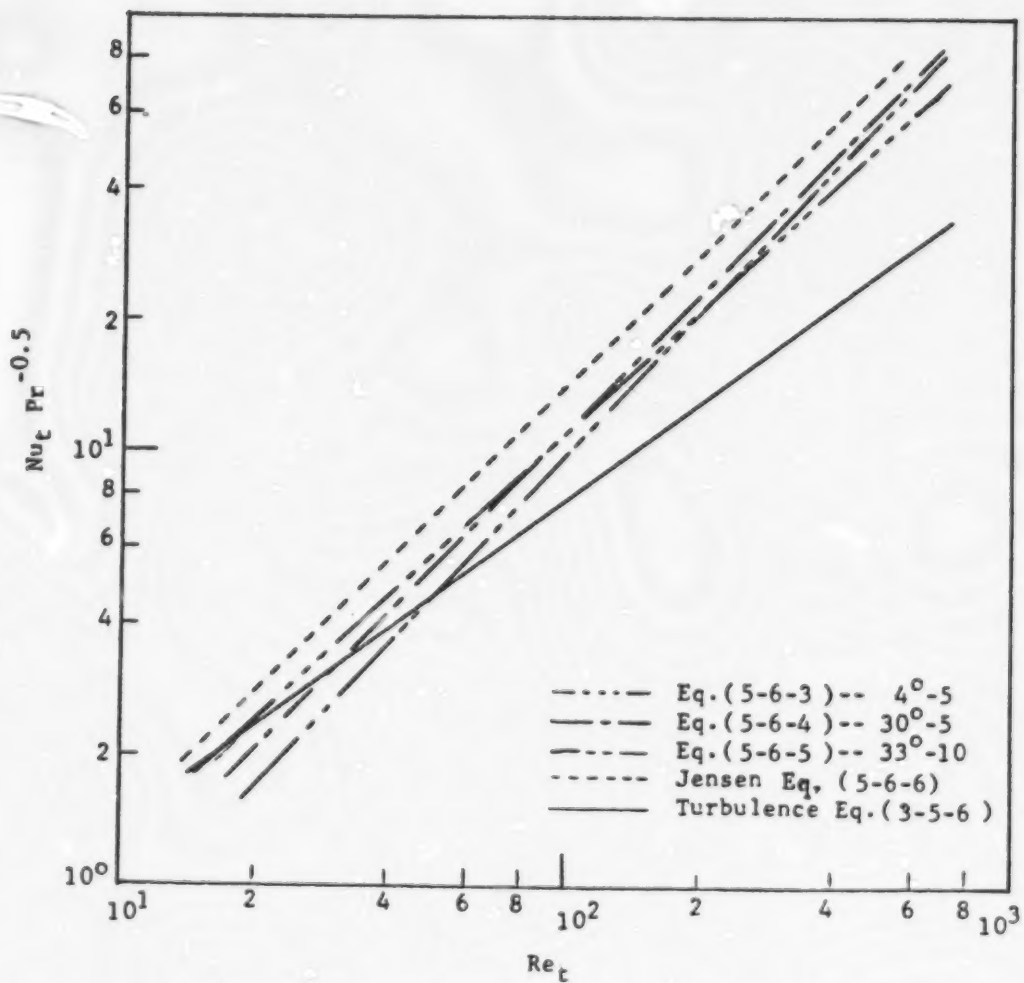


Fig. 34 Comparison with other equations for turbulence-centered model

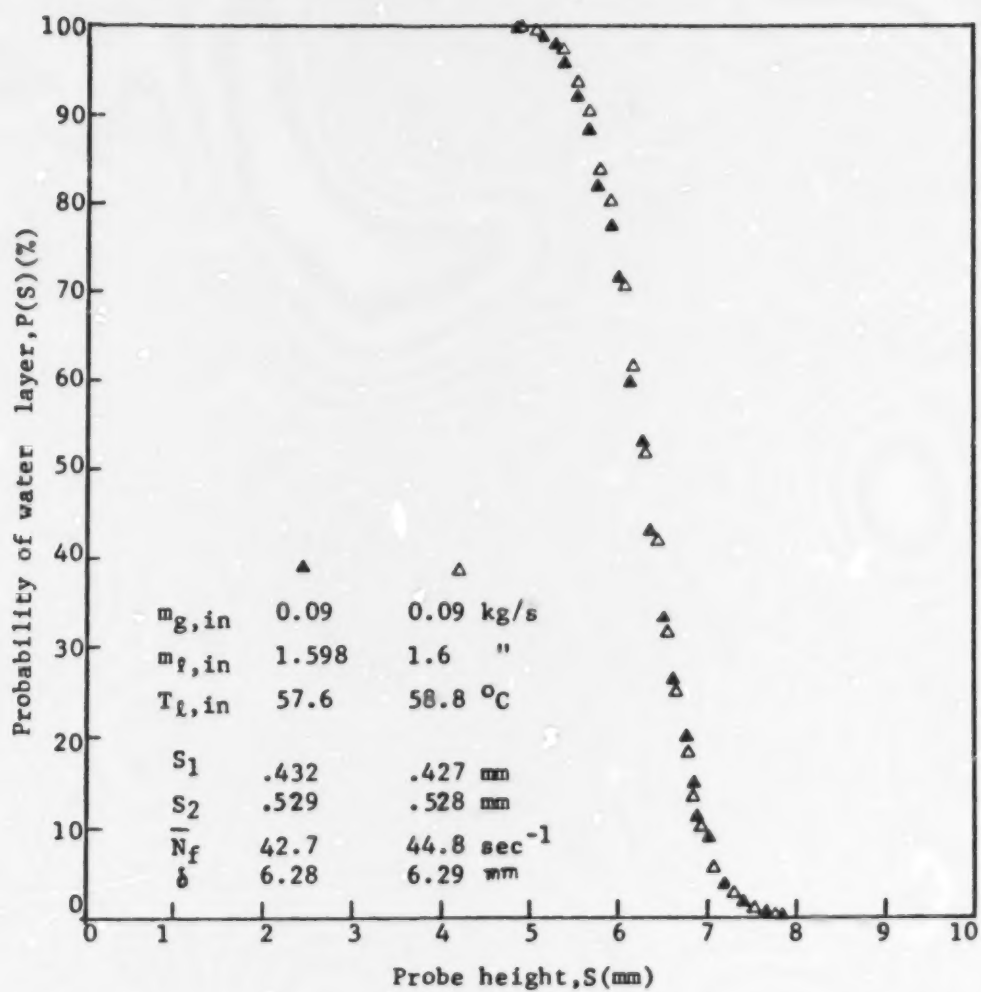


Fig. 35 Repeat of experiment to show reproducibility of data for interfacial wave structure

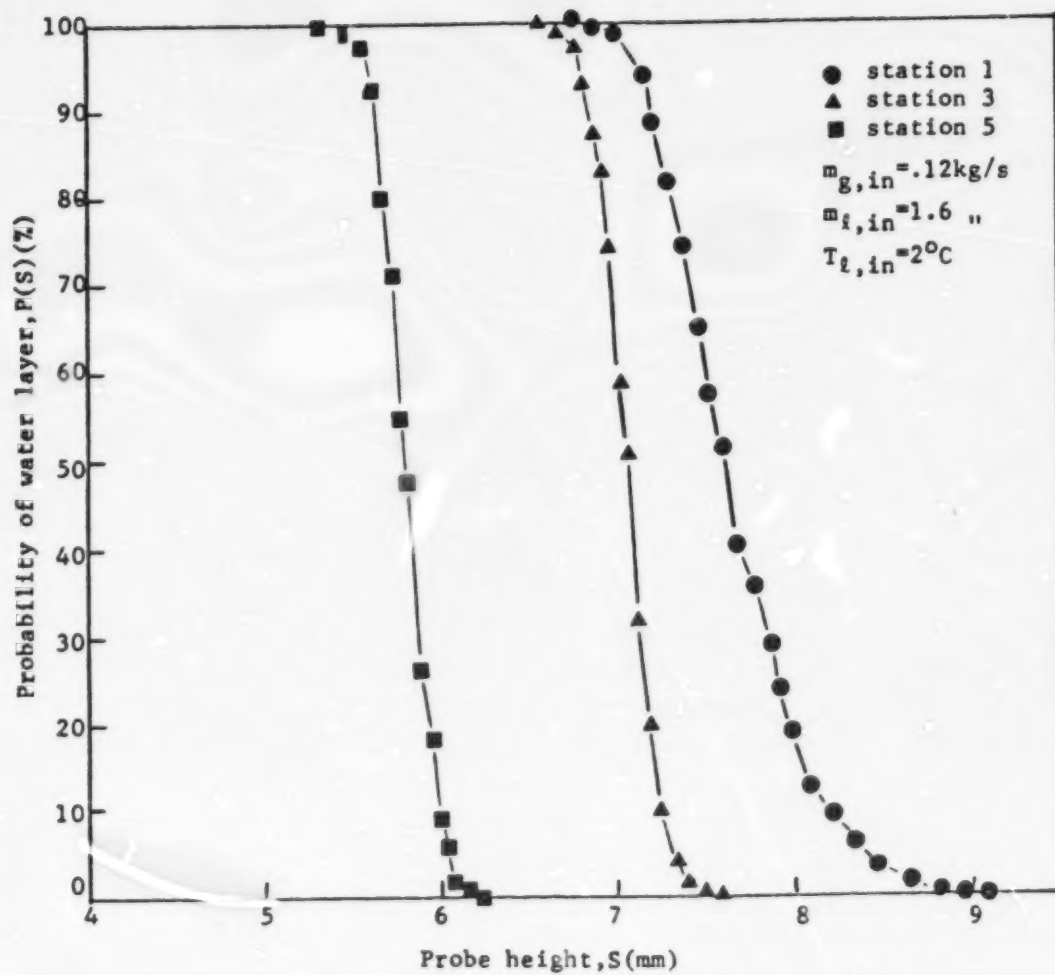


Fig. 36 Probability distributions at different stations

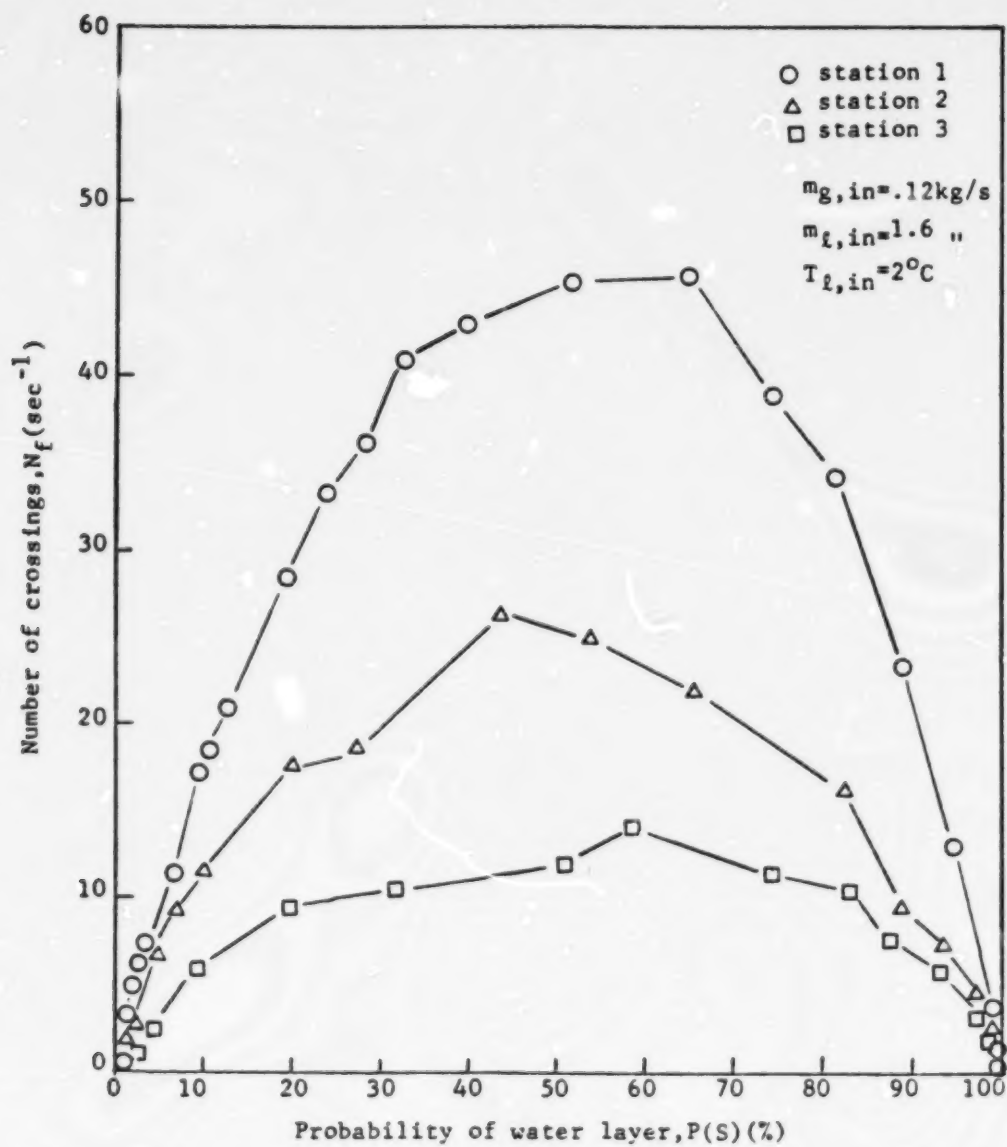


Fig. 37 Distribution of frequency of crossings at different stations

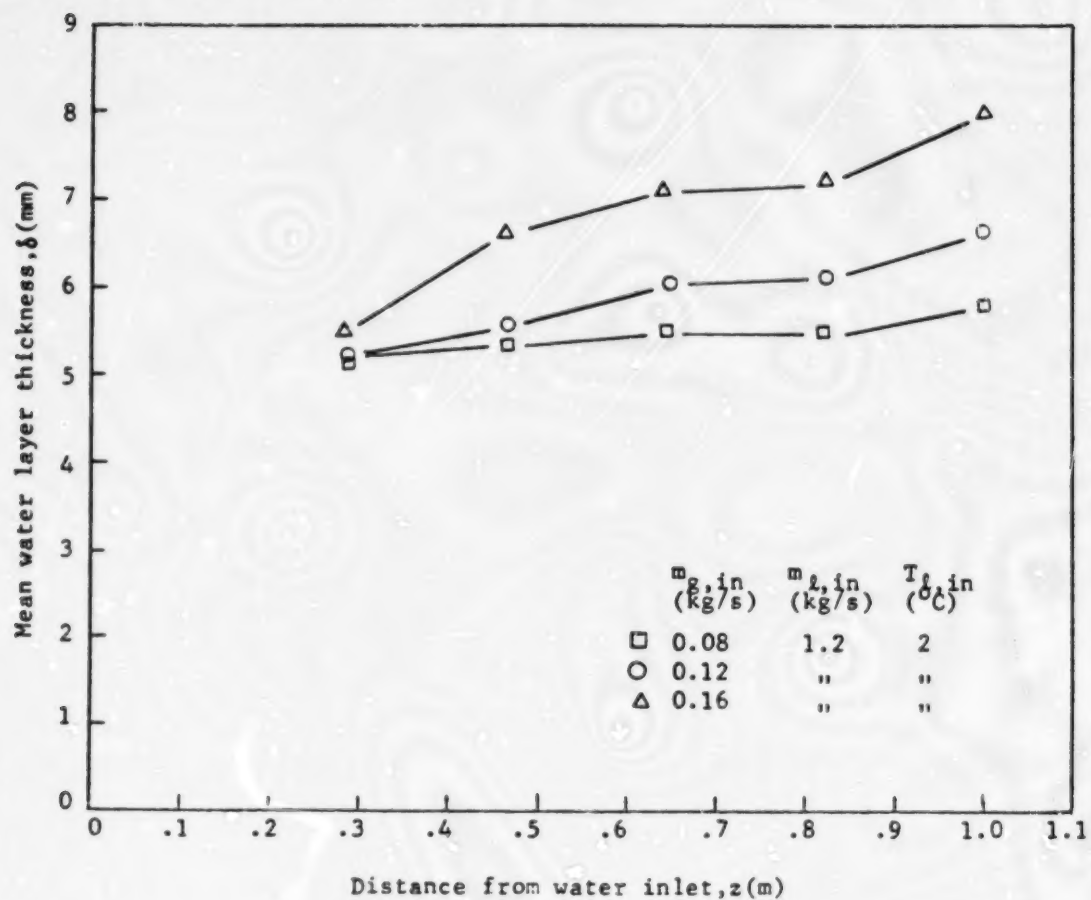


Fig. 38 Axial mean water layer thickness profile as a function of inlet steam flow rate

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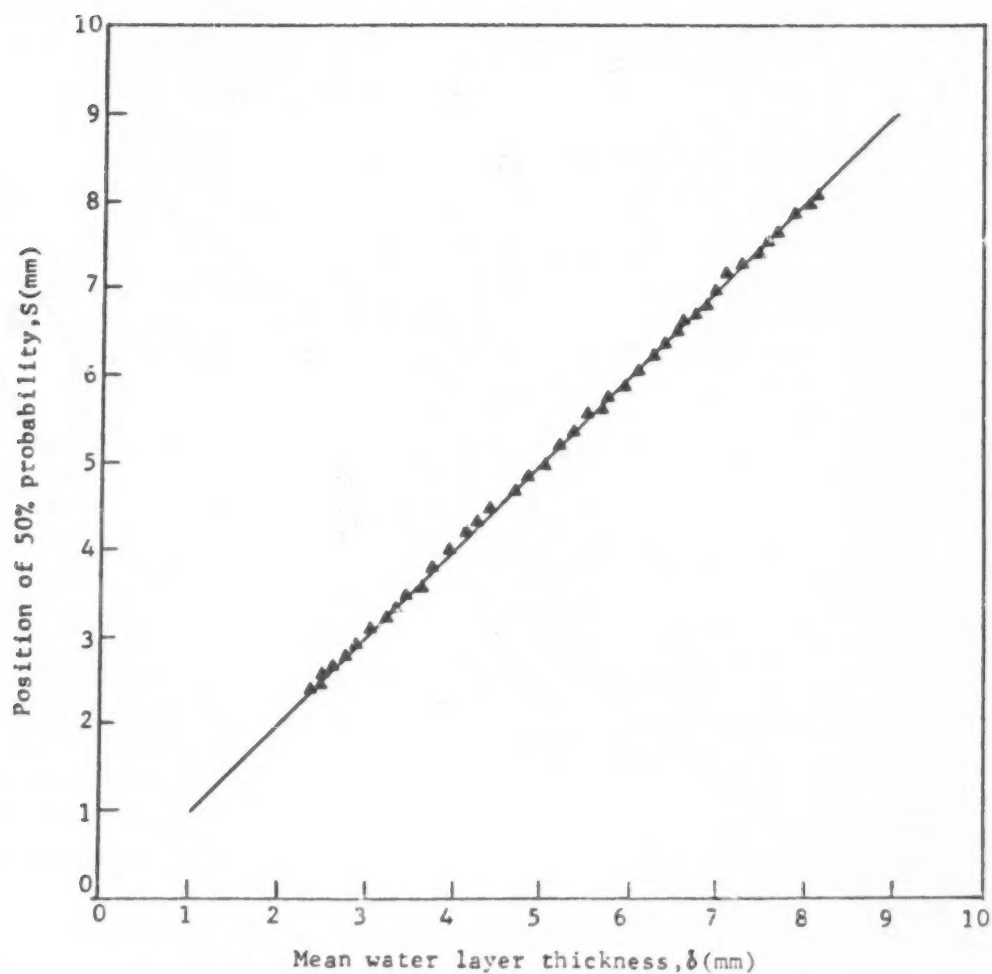


Fig. 39 Comparison of mean water layer thickness with the thickness with 50% probability

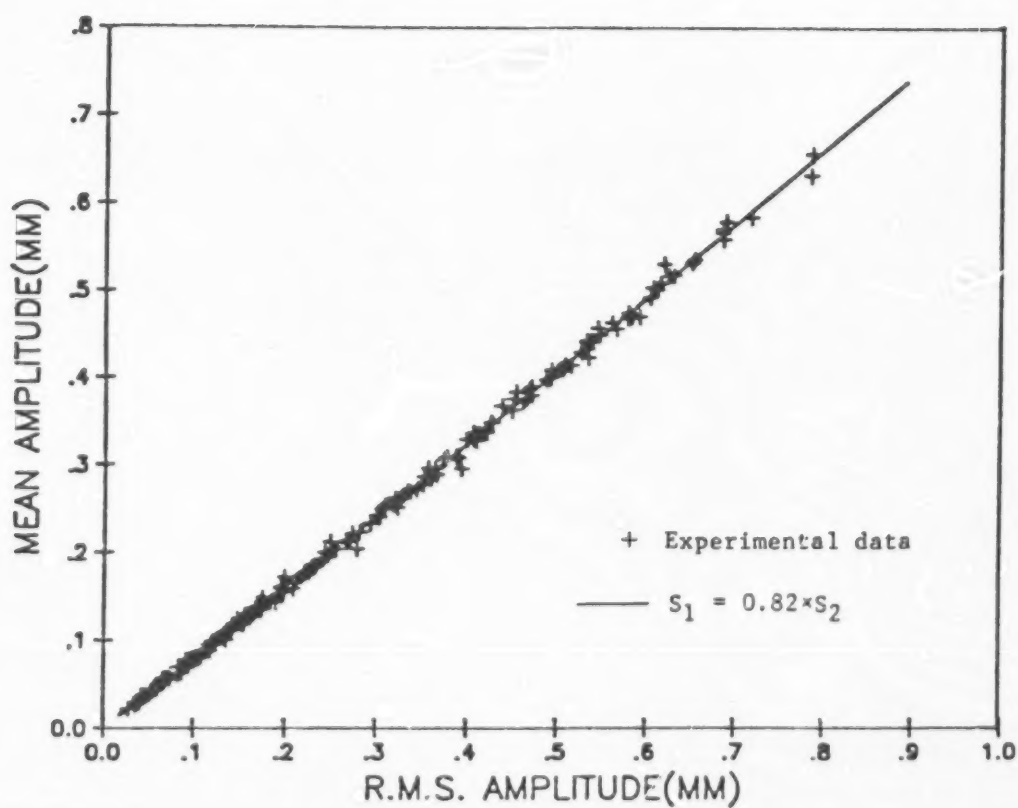


Fig. 40 Relation between mean wave amplitude and r.m.s. wave amplitude

|                                                                                                                                                                                                                                                                                                                                                                                                                       |  |                                                                              |  |                                                            |  |
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